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RESEARCH REPORT

STUDY OF UNCONVENTIONAL THERMAL, MECHANICAL, AND NUCLEAR LOW-POLLUTION-POTENTIAL POWER SOURCES FOR URBAN VEHICLES

to

UNITED STATES DEPARTMENT OF HEALTH,
EDUCATION AND WELFARE
NATIONAL CENTER FOR AIR
POLLUTION CONTROL

March 15, 1968

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SUMMARY REPORT

on

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UNITED STATES DEPARTMENT OF HEALTH,
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March 15, 1968

by

J. A. Hoess, E. S. Cheaney, F. A. Creswick,
D. A. Trayser, R. D. Fischer, A. B. Timberlake,
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STUDY OF UNCONVENTIONAL THERMAL, MECHANICAL, AND NUCLEAR LOW-POLLUTION-POTENTIAL POWER SOURCES FOR URBAN VEHICLES

by

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INTRODUCTION

This report presents the results of a study conducted by Battelle - Columbus Laboratories (BCL) for the National Center for Air Pollution Control (NCAPC), to investigate unconventional thermal, mechanical, and nuclear low-pollution-potential power sources for urban vehicles. The study was conducted during the period of May 17, 1967, through March 15, 1968.

The purpose of the program was to provide background information and recommendations required by NCAPC for development of future research and development programs on low-pollution-potential propulsion systems for urban vehicles. The study itself consisted of a first screening of possible alternative systems to identify those that were the most promising so that future efforts could be concentrated on these. So as not to screen out systems that could be enhanced through directed research and development efforts, alternative systems were evaluated on the basis of the characteristics that they could be expected to have 10 to 15 years in the future, assuming that development efforts were expended on them. As a result of the study, a few alternative propulsion systems have been identified as being quite promising for urban-vehicle application, while a large number of systems were found to be unsuitable for such application.

Though the scope of the project was broad, there were a number of notable power sources not included, viz., Diesel - and Otto-cycle, internal-combustion engines along with electrochemical systems and electric drive trains, all of which were being investigated elsewhere. NCAPC has a continuing program with respect to devices to control the emissions from Diesel - and Otto-cycle engines. The investigation of electrochemical systems and electric drive trains was to be conducted by Arthur D. Little, Inc. (ADL).

Figure 1, showing an alternatives network, illustrates in more detail the various alternative devices and systems investigated during this program and their relationships with one another. In Figure 1, any horizontal path represents a specific form of energy. A given propulsion system can be described by starting from the appropriate energy-storage means and tracing a path downward along vertical paths and either left or right along the horizontal energy paths. For example, recent, experimental, gas-turbine-powered automobiles have a propulsion system that is described by starting at the chemical-fuel energy source, moving to the left along the chemical-energy path and down through open Brayton-cycle (gas turbine) engines, proceeding to the right along the upper mechanical-energy path and down through mechanical transmissions, and

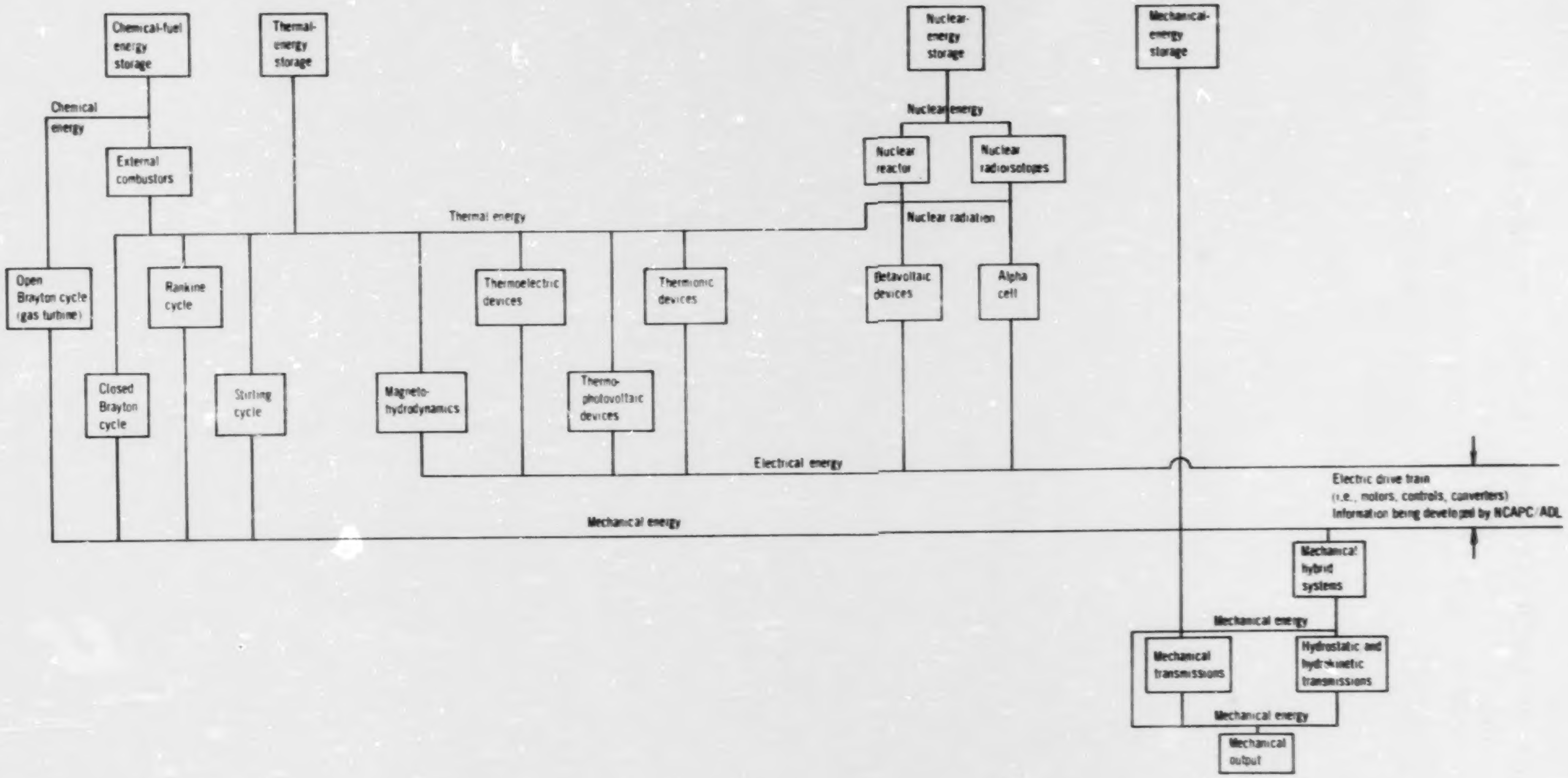
Chemical-fuel energy storage
Thermal-energy storage
Nuclear-energy storage
Mechanical-energy storage
Chemical energy
External combustors
Nuclear energy
Nuclear reactor
Nuclear radioisotopes
Thermal energy
Nuclear radiation
Open Brayton cycle (gas turbine)
Rankine cycle
Closed Brayton cycle
Stirling cycle
Magneto-hydrodynamics
Thermoelectric devices
Thermo-photovoltaic devices
Thermionic devices
Betavoltaic devices
Alpha cell
Electrical energy
Mechanical energy
Electric drive train
(i.e., motors, controls, converters)
Information being developed by NCAPC/ADL
Mechanical hybrid systems
Mechanical energy
Mechanical transmissions
Hydrostatic and hydrokinetic transmissions
Mechanical energy
Mechanical output

FIGURE 1. ALTERNATIVES NETWORK FOR STUDY OF UNCONVENTIONAL THERMAL, MECHANICAL, AND NUCLEAR LOW-POLLUTION-POTENTIAL POWER SOURCES FOR VEHICLES

finally proceeding to a mechanical form of energy output. Likewise, numerous other combinations of components comprising potential vehicle-propulsion systems can be traced on Figure 1.

The mechanical form of output energy involves the rotating of a shaft under torque. This output would be compatible with vehicular systems deriving their propulsive force through traction wheels or propellers. Other methods for propelling vehicles are available, such as linear electric motors, jet engines, rockets, pneumatics, etc., but they are either electrical devices or are considered as very unlikely alternatives for urban vehicles.

Mechanical hybrid systems refer to propulsion systems that incorporate both a mechanical-energy-storage device and other nonmechanical-energy-storage and -conversion devices.

CONCLUSIONS

The following conclusions have resulted from the studies and investigations conducted during this research program.

(1) Of the alternative system and devices falling within the scope of this program, the most attractive propulsion systems for urban vehicle application are those including Rankine-cycle (steam) engines, Brayton-cycle (gas turbine) engines, and Stirling-cycle (hot air) engines. Further, Stirling engines are not as attractive for general automotive application as Rankine (steam) engines and gas turbines. The superior efficiency of the Stirling engine, however, may make it an attractive alternative for city bus application.

(2) Nuclear devices, pure mechanical systems, thermal-energy storage, and direct thermal-to-electric energy converters are not attractive for urban-vehicle application.

(3) External combustors such as those used on Rankine (steam) and Stirling engines have excellent characteristics with respect to the emission of carbon monoxide (CO) and unburned hydrocarbons (HC). They display considerable variation, however, in the emission of oxides of nitrogen (NO_x). The mass-emission-rates (mass of emissions per unit energy of fuel burned) of CO and HC for external combustors is generally less than 4 percent of that of uncontrolled-emission gasoline engines. Leaded fuels are not used, so emission of compounds of lead is not a problem.

(4) Gas turbine engines, which have combustors that are very similar to external combustors, also display good characteristics with respect to the emission of CO and HC and display considerable variation in the emission of NO_x. At least one present-day turbine has been tested with mass-emission rates of CO and HC that are only 4 to 5 percent of those typical for uncontrolled-emission gasoline engines.

(5) Because of weight and size requirements of the fuel plus appropriate container, the unconventional chemical fuels investigated, with the possible exception of propane and butane, are not attractive for general use in urban vehicles. The principal exception to this would be the very low energy vehicle applications. To overcome their weight and size handicap, unconventional fuels, such as ammonia, methane and hydrogen, would have to possess emission characteristics significantly better than those of conventional fuels before they could be considered as legitimate contenders for urban-vehicle application. When considering external combustor systems, it is believed unlikely that these latter unconventional fuels will possess sufficiently improved emission characteristics over conventional fuels, particularly with respect to NO_x, to compensate for their weight and size handicap.

(6) While a pure mechanical-energy-storage system is not attractive for urban-vehicle application, a flywheel-engine hybrid system may be desirable for bus application. In such a system, the flywheel would store only enough energy to assist in acceleration and would be "recharged" between start-ups.

RECOMMENDATIONS

Future study and research efforts on the systems and devices falling within the scope of this program should be concentrated on the following subjects:

(1) Development of criteria for the design of external combustors suitable for automotive application and having minimum emission of NO_x. This work should include determination of the minimum emission levels applicable to both conventional and unconventional fuels. Only if unconventional fuels show considerable advantage with respect to emissions should future efforts be devoted to detailed studies of their production, distribution, cost, and optimum container characteristics.

(2) Development of an advanced technology, Rankine-cycle (steam) engine.

(3) Studies and research to minimize the emissions of NO_x from gas turbines. This work should be closely coordinated with the combustor work listed uner Item (1) above.

(4) Limited studies and research in connection with gas turbine engines and Stirling engines.

(5) Further study and possible development of a flywheel-engine hybrid system for bus application.

It should be noted that the above-recommended subjects for future study and research have grown out of one part of a three-part planning study being conducted by NCAPC. As discussed in the Introduction, NCAPC has a continuing program with respect to Diesel- and Otto-cycle engines and ADL is studying electrochemical systems and electric drive trains. Final decisions as to the most desirable subjects for research and the setting of priorities for such research will result from an evaluation of the most attractive systems identified in each of the three study areas.

TECHNICAL DISCUSSION ORGANIZATION AND TERMINOLOGY

In presenting the results of the study, it has been found convenient to divide the technical discussion into two major sections. The first, Evaluation of Alternative Systems, contains a discussion of the evaluation methodology used, system evaluation criteria, a summary of projected "1980" system characteristics, and the results of system evaluation. The second section, Present and Projected State of the Art for Alternative Systems, contains detailed discussions of the present and projected physical, performance, and cost characteristics for all of the alternative systems and devices investigated during the program.

One last point has to do with terminology. The engine that is presently used in almost all automobiles is, technically speaking, an Otto-cycle engine, but it is more commonly referred to as a gasoline engine, spark-ignition engine, or internal-combustion engine. It is believed that the majority of the readers of this report will be most accustomed to thinking with the term "gasoline engine"; thus, this term is used to designate Otto-cycle engines in the body of the report. Diesel-cycle engines are commonly referred to as diesel engines and this terminology should not cause any confusion. The term internal-combustion (IC) engines is used to refer to both gasoline and diesel engines. Finally, throughout this report, many different forms of energy and power are discussed. To avoid the possibility of misinterpretation, the notations (t) thermal, (s) shaft, or (e) electrical are added to energy or power designations so as to define the specific form of the energy or power being referred to.

EVALUATION OF ALTERNATIVE SYSTEMS

- SYSTEM EVALUATION METHODOLOGY
- SYSTEM EVALUATION CRITERIA
- SUMMARY OF PROJECTED "1980" CHARACTERISTICS FOR ALTERNATIVE SYSTEMS
- SYSTEM EVALUATION

EVALUATION OF ALTERNATIVE SYSTEMS

SYSTEM-EVALUATION METHODOLOGY

The particular evaluation procedure used in the study was devised both to rank order the alternative systems and devices investigated with respect to their relative attractiveness as urban vehicle propulsion systems and to identify the obviously unacceptable alternatives so they could be excluded from further research efforts. This procedure consisted of the following principal steps. First, six specific urban-vehicle missions or applications were hypothesized. Then, both vehicle specifications and minimum-allowable performance, or "threshold", propulsion-system specifications were hypothesized for each of the vehicle applications in question.

Concurrent with the above, state-of-the-art studies of all of the alternative systems and components of interest were conducted. These studies were not only concerned with documenting the past and present state of the art, but had as their primary objective the provision of the basic information and knowledge required to project the characteristics that the systems of interest might have 10 to 15 years in the future, i.e., "1980" characteristics, assuming that sizable development efforts were expended on them. Such projection of system characteristics is required both to assist in the identification of desired topics for future research and to insure that advances that can be expected to result from future directed research are suitably factored into the evaluation of the alternative systems.

The "1980" characteristics projected for any specific system represent the summary judgment of researchers who are knowledgeable with respect to the pertinent state of the art and who "qualified" themselves for making such projections or forecasts by conducting the particular state-of-the-art studies in question. These summary judgments were arrived at through the process of engineering-design visualization wherein knowledge of parameter trade-off possibilities, gap areas of knowledge, developmental problems, and possible technological advances are incorporated into an overall assessment of how the component could be expected to be advanced through future research. In some instances this may merely result in an extrapolation of current technological trends and in others it may not. For example, in the case of gas turbines for passenger cars, only minor improvements are projected for engine specific weight and volume. While future technological advances could be expected to reduce the values of these parameters, it is believed that such advances will be utilized in a manner to reduce engine initial cost rather than size or weight, as initial cost is the largest current problem for passenger-car gas turbines.

The discussions contained in the section Present and Projected State of the Art for Alternative Systems attempt to elucidate as much as possible the many considerations entering into the projection of the "1980" component characteristics.

Finally, by comparing the characteristics projected for each of the alternative systems against the hypothesized threshold propulsion-system specifications, the obviously unacceptable propulsion systems were identified and the relative attractiveness of those systems "above threshold" was determined.

SYSTEM EVALUATION CRITERIA

Table 1 lists the six vehicles of interest, along with the hypothesized vehicle and threshold propulsion-system specifications. The vehicle specifications and the propulsion-system power, energy, and weight specifications were developed principally by ADL personnel in cooperation with NCAPC and Battelle personnel and have been mutually agreed upon. The development of these specifications will be discussed elsewhere and thus will not be repeated here.

Since the scope and thus the characteristics of the alternative systems investigated during this particular program varied so widely, however, it was thought desirable to also develop propulsion-system volume, initial cost, and start-up-time specifications to assist in the evaluation of these alternatives. The open literature contains a number of propulsion-system-criteria analyses based on permissible component weight, but similar detailed information involving the parameters of volume and cost could not be found. In view of this data gap, it became necessary to conduct a cursory propulsion-system volume and cost analysis for a range of current American vehicles. The vehicles selected for this study included four passenger cars, two delivery vans, and one city bus. A discussion of the assumptions and development of the resulting volume and cost specifications is included in Appendix A.

To give due cognizance to the impatience of the American motorist, the absolute maximum or "threshold" specification for time to self-sustaining engine operation is estimated at 1 minute for the personal vehicles and 3 minutes for the commercial vehicles. Twice this amount of time is considered a threshold specification for full power output. Of course, it would be very desirable to have start-up times of 30 seconds or less; however, as a threshold specification, the longer times listed above are believed to be more appropriate.

SUMMARY OF PROJECTED "1980" CHARACTERISTICS FOR ALTERNATIVE SYSTEMS

Table 2 contains a summary of the projected "1980" propulsion-system-component characteristics developed during the state-of-the-art studies of individual components. As previously mentioned, detailed discussions of pertinent background information and the process of projecting these characteristics are presented in the section Present and Projected State of the Art for Alternative Systems.

The characteristic values listed in Table 2 for heat engines and direct converters are applicable to automobile-type duty cycles, life requirements, and production quantities. The specific weight, volume, and cost values would be somewhat higher for bus-type application. Also, the listed costs are intended to reflect manufacturer's minimum selling price (i.e., dealer's cost at the manufacturer's plant).

By inspection of the "1980" component characteristics listed in Table 2 and the threshold specifications for urban-vehicle propulsion systems listed in Table 1, it is quite evident that magnetohydrodynamics and nuclear reactors and radioisotopes cannot meet threshold specifications with respect to one or more characteristics. This also

TABLE 1. HYPOTHESIZED VEHICLE AND THRESHOLD PROPULSION-SYSTEM SPECIFICATIONS

	Family Car	Commuter Car	Utility Car	Delivery Van	City Taxi	City Bus
Assumptions						
Acceleration to, mph	60	60	30	40	40	30
in, sec	15	30	10	20	15	15
Range[a], miles	200	50	50	60	150	120
Seats or Payload, lbs	6	4	2	2500	6	10,000
Loaded Weight, lb	4000	2500	1700	7000	4000	30,000
Curb Weight, lb	3500	2100	1400	4500	3500	20,000
Weight[b], lb, Assignable to All Propulsion-System Components, Including Energy Storage, Controls, Etc.	1500	900	600	1700	1500	6,000
Volume[b], ft^3, Assignable to All Propulsion-System Components	28	20	16	28	28	175
Initial Cost[c], $, Assignable to Propulsion System	820	670	600	860	820	8,100
Start-up Time to Power Output, min	1	1	1	3	3	3
Frontal Area, ft^2	25	18	18	42	25	80
Drag Coefficient	0.35	0.25	0.25	0.85	0.35	0.85
Derived Parameters[d]						
Maximum Delivered Power[e], hp(s)	94	30	16	96	48	180
Cruising Power[f], hp(s)	47	15	8	48	24	90
Maximum Cruising Velocity, mph	75	57	47	48	55	38
Maximum Attainable Velocity, mph	100	80	65	64	77	55
Delivered Energy[a], hp(s)-hr	134	13.4	10.7	60	100	400

(a) Using NCAPC cycle for all vehicles except the city bus for which the ADL cycle is used.

(b) Assuming a more or less integral package propulsion system similar to present systems.

(c) Assuming:

- Manufacturer's minimum selling price (i.e., dealer's cost at manufacturer's plant)
- Delivered energy costs smiliar to present delievered energy costs [i.e., $/hp(s)-hr].

(d) Derived power and energy parameters apply to net propulsion-system output power and energy after accounting for all propulsion-system and driveline losses.

(e) Peak power required for acceleration.

(f) Cruising power is 50 percent of the maximum power. Maximum power is required for only short periods of time.

TABLE 2. SUMMARY OF PROJECTED "1985" CHARACTERISTICS FOR VEHICLE PROPULSION-SYSTEM COMPONENTS

Component	Specific Weight	Specific Volume	Maximum Efficiency, percent	Start-Up Time, sec	Specific Cost(a)
Heat Engines(b)	lb/hp(t)	ft³/hp(t)			$/hp(t)
Gas turbine(c)	3-4	0.09-0.12	35-30	<15	3-4.5
Rankine (steam) engine	5-8	0.12-0.16	30-25	<15	4-6
Stirling engine	8-10	0.15-0.20	40-35	<15	5-7
Direct Thermal-to-Electric Energy Converters(b)	lb/hp(e)	ft³/hp(e)			$/hp(e)
Magnetohydrodynamic	(lower limit for useful MHD) (considered to be 1000 hp(e))				40-110
Thermoelectric	13	--	10	--	Extremely high
Thermionic	10	0.2	10	<3.5 min	Very high [i.e., $100/hp(e)]
Thermophotovoltaic	23	--	13	--	--
Chemical-Fuel-Energy Storage	lb/hp(t)-hr(d)	ft³/hp(t)-hr(d)			$/hp(t)-hr(e)
Conventional Fuels	0.15	0.0029			0.004
Propane (LPG)	0.22	0.0042			0.008
Ammonia	0.31	0.0150			0.013
Methane (cryogenic)	0.27	0.0180			0.003
Hydrogen (cryogenic)	0.44	0.0440			0.010
Hydrogen (magnesium hydride)	0.74	0.012			0.004-0.010 (0.33)(f)
Thermal-Energy Storage					
Lithium fluoride	3.30	0.032			--
Mechanical-Energy Storage	lb/hp(t)-hr	ft³/hp(t)-hr			$/hp(t)-hr
Flywheel	100	0.25			(100)(f)
Rubber Spring	140	2.00			--
Hydraulic Accumulator	300	7.90			--
Nuclear Systems					
Reactors	An appropriately shielded reactor system for urban-vehicle application would: have a specific weight greater than 250 lb/hp(e) or hp(t); be extremely large; require hours to start; and cost thousands of dollars per hp(e) or hp(t).				
Radioisotopes	Radioisotope thermal power generators could probably be developed to have acceptable weight and size characteristics for urban-vehicle application. Their high cost, which is currently in the tens of thousands of dollars per horsepower (t), and unavailability, however, make them unfeasible for this application in the foreseeable future. Also, radioisotopes cannot be "turned off" and would continually emit heat at their design capacity.				
Vehicle Transmissions(g)	lb/hp(t)	ft³/hp(t)			$/hp(t)
Mechanical (Including Clutch)					
Automobile	0.5	0.004	95		0.4
Truck	2.0	0.020	95		6.0
Hydrokinetic					
Automobile	0.7	0.005	90		1.0
Truck	3.0	0.030	90		12.0
Hydrostatic					
Automobile	--	--	80		--
Industrial	2-3	0.02-0.04	80		8-20

Footnotes appear on following page.

Footnotes for Table 2

(a) The listed costs are intended to reflect manufacturer's minimum selling price (i.e., dealer's cost at the manufacturer's plant). While all of the values listed in this table represent best estimates of mean values for given parameters, the values listed for specific cost are based on a minimum of information and the largest number of intangible factors and are, therefore, subject to the greatest variation.

(b) Characteristic values listed for heat engines and direct converters are applicable to automobile-type duty cycles, life requirements, and production quantities. The weight, volume, and cost values must be increased somewhat for bus-type application.

(c) It is not believed that a competitive vehicular gas turbine could be developed for the utility car [i.e., 16 hp(s)] application.

(d) Specific weight and volume estimates for energy storage include the weight and volume of both the storage medium itself and an appropriate container.

(e) The listed specific fuel costs are for the fuel only and are based on current bulk costs less road taxes. The cost of some of the lesser used unconventional fuels could be expected to be reduced by increased usage.

(f) Estimated specific initial cost for rechargeable, energy storage devices.

(g) Listed values for vehicle transmissions represent the current state of the art. These mechanisms are highly developed at present, and future improvements will be evolutionary in nature. The principal exception to this is the case of high-power, high-pressure, ultrahigh-speed hydrostatic transmissions.

applies to alpha- and betavoltaic cells which would use radioisotopes as their energy source. Thus, these devices were determined to be unsuitable for urban-vehicle application and were not subjected to further evaluation.

Estimates of propulsion-system characteristics for a wide variety of alternative systems are given in Tables 3 through 6. The estimated "absolute" system characteristics given in these tables are based on the projected "specific" component characteristics given in Table 2.

Table 3 shows a sample calculation for the estimation of propulsion-system characteristics for the family-car application. The calculations for the other five vehicle applications are similar to those illustrated in Table 3, and the calculation sheets for these are contained in Appendix B. Table 4 is a summary of the estimated propulsion-system weights, volumes, and initial costs calculated for all six vehicle applications. Table 5 shows the variation in estimated Rankine (steam)-engine propulsion-system weight and volume for a selection of chemical-fuel and thermal-energy storage means. Finally, Table 6 lists the assumptions and calculations used to estimate flywheel and associated transmission weights, volumes, and initial costs for flywheel-engine hybrid propulsion systems for the family-car and city-bus applications.

While the procedure illustrated in Table 3 for estimating propulsion-system characteristics is an oversimplification of the complexities evolved in matching engines and transmissions to vehicles, it is adequate for the purposes of this study. Maximum engine power is obtained by assuming a transmission and driveline efficiency and applying this to the maximum power to be delivered to the vehicle, which is specified in Table 1. The listed efficiencies are as high as would be expected when assuming that: a basically mechanical transmission and driveline are used with the gas turbine; the Rankine (steam) engine requires little or no transmission provisions; the Stirling engine uses a hydrokinetic transmission; a hydrostatic transmission is used with the flywheel; and the direct converters have an electric drive system.

Rated engine power for the Rankine engine is reduced 20 percent below maximum power because the Rankine engine would have a power-surge capability equivalent to at least 25 percent of its rated power. The flywheel system characteristics listed apply to a "pure" system. That is, one in which all of the energy required for the vehicle application in question is stored in the flywheel. The direct, thermal-to-electric, energy converters are assumed to be part of a "hybrid" system along with suitable batteries, motors, and controls. Thus, their rated power is assumed to be equivalent to that required for cruising rather than to maximum engine power.

The specific weights, volumes, and costs and peak engine efficiencies listed for the heat engines in Table 3 and for the other vehicle applications, whose calculation sheets are contained in Appendix B, were arrived at by assuming that the most attractive values listed in Table 2 are applicable to the vehicle application requiring the highest power engine. Likewise, the least attractive values in Table 2 were assigned to the application requiring the lowest power engine. Characteristic values for the intermediate-power applications were then obtained by assuming that they would vary between these two extreme points in a logarithmic manner.

Since the characteristic values listed in Table 2 for the heat engines and direct converters are applicable to automobile-type duty cycles, life requirements, and production qualities, it was necessary to increase these values somewhat for the bus application. This is necessary, of course, because a bus propulsion system must be heavy

TABLE 3. ESTIMATED PROPULSION-SYSTEM CHARACTERISTICS FOR FAMILY-CAR APPLICATION

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermoelectric Converter[a]	Thermionic Converter[a]	Thermophotovoltaic Converter[a]	Magnetohydrodynamics and Nuclear Devices
Maximum Delivered Power, hp(s)	94	94	94	94	94	94	94	Found to be unsuitable for this application - see discussion.
Assumed Transmission and Drive Efficiency, %	90	95	85	80	80	80	80	
Maximum Engine Power, hp(s) or hp(e)	104	99	110	117	117	117	117	
Rated Engine Power, hp(s) or hp(e)	104	79	110	117	59	59	59	
Specific Engine Weight, lb/hp(s) or hp(e)	3.3	5.7	8.4	--	13	10	13	
Engine Weight, lb	340	450	920	--	770	590	770	
Specific Transmission Weight, lb/hp(s)	0.5	0.0	0.7	2.0	--	--	--	
Transmission Weight, lb	52	0.0	77	230	Weight of battery, motors and controls			
Energy Delivered to Vehicle, hp(s)-hr	134	134	134	134	134	134	134	
Energy Delivered by Engine, hp(s)-hr or hp(e)-hr	149	141	158	--	168	168	168	
Peak Engine Efficiency, %	33	29	38	--	10	10	13	
Assumed Average Efficiency, %	23	20.5	26.5	80	9	9	11.5	
Required Stored Energy, hp(t)-hr	649	690	596	168[b]	1870	1870	1460	
Specific Weight, Conventional Fuel, lb/hp(t)-hr	0.15	0.15	0.15	100[c]	0.15	0.15	0.15	
Energy Storage Weight, lb	97	100	90	16,800	280	280	220	
Engine, Transmission, and Conventional Fuel Weight, lb	490	550	1100	17,000	1000 +	900 +	1000 +	
Specific Engine Volume, ft^3/hp(s) or hp(e)	0.10	0.13	0.16	--	--	0.2	--	
Engine Volume, ft^3	10.4	10.3	17.6	--	--	11.8	--	
Specific Transmission Volume, ft^3/hp(s)	0.004	0.0	0.005	0.02	--	--	--	
Transmission Volume, ft^3	0.42	0.0	0.55	2.34	Volume of batteries, motors and controls			
Specific Volume Conventional Fuel, ft^3/hp(t)-hr	0.0029	0.0029	0.0029	0.25[d]	0.0029	0.0029	0.0029	
Energy Storage Volume, ft^3	1.88	2.00	1.73	42.0	5.41	5.41	4.24	
Engine, Transmission, and Conventional Fuel Volume, ft^3	13	12	20	44	Similar to thermionic	17 +	Similar to thermionic	
Specific Engine Cost, $/hp(s) or hp(e)	3.4	4.4	5.4	--	Not known but is expected to be $100 $/hp(e) or higher			
Engine Cost, $	350	350	600	--				
Specific Transmission Cost, $/hp(s)	0.4	0.0	1.0	3.00	--	--	--	
Transmission Cost, $	42	0.0	110	350	Cost of batteries, motors, and controls			
Specific Energy Container Cost, $/hp(t)-hr	Nominal	Nominal	Nominal	100[e]	Nominal	Nominal	Nominal	
Energy Container Cost, $	--	--	--	16,800	--	--	--	
Engine, Transmission, and Energy Container, Cost, $	390	350	710	17,000		Very high		

[a] Considered to be part of a hybrid system. Their power rating is based on the cruise power requirement which is one-half the maximum power. The weight, size, and cost of associated batteries, motors, and controls would have to be added to the values listed in this table for the converters alone.

[b] Value given in hp(s)-hr.

[c] Value given in lb/hp(s)-hr.

[d] Value given in ft^3/hp(s)-hr.

[e] Value given in $/hp(s)-hr.

TABLE 4. SUMMARY OF ESTIMATED PROPULSION-SYSTEM CHARACTERISTICS FOR SIX URBAN-VEHICLE APPLICATIONS

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter[(b)]	Thermionic Converter[(b)]	Thermo-photo-voltaic Converter[(b)]	Magneto-hydrodynamics and Nuclear Devices
				Family Car				
Engine, Transmission, and Fuel Weight, lb	490	550	1100	17,000[(a)]	1,000+	900+	1,000+	Found to be unsuitable for any of these applications - see discussion.
Engine, Transmission, and Fuel Volume, ft^3	13	12	20	44[(a)]	--	17+	--	
Engine and Transmission Cost, $	390	350	710	17,000[(a)]		Very high[(a)]		
				Commuter Car				
Engine, Transmission, and Fuel Weight, lb	160	190	360	1,800[(a)]	280+	220+	270+	
Engine, Transmission, and Fuel Volume, ft^3	4.2	4.0	6.7	4.9	--	4.3+	--	
Engine and Transmission Cost, $	160	140	260	1,800[(a)]		Very high[(a)]		
				Utility Car				
Engine, Transmission, and Fuel Weight, lb	(not applicable)	120	210	1,400[(a)]	150+	120+	150+	
Engine, Transmission, and Fuel Volume, ft^3		2.4	4.0	3.8	--	2.4+	--	
Engine and Transmission Cost, $		80	150	1,400[(a)]		Very high[(a)]		
				Delivery Van				
Engine, Transmission, and Fuel Weight, lb	450	500	1100	7,700[(a)]	900+	720+	880+	
Engine, Transmission, and Fuel Volume, ft^3	12	11	19	21[(a)]	--	14+	--	
Engine and Transmission Cost, $	400	360	720	7,900[(a)]		Very high[(a)]		
				City Taxi				
Engine, Transmission, and Fuel Weight, lb	300	340	630	13,000[(a)]	600+	510+	550+	
Engine, Transmission, and Fuel Volume, ft^3	7.6	7.2	12	32[(a)]	--	10+	--	
Engine and Transmission Cost, $	230	200	400	13,000[(a)]		Very high[(a)]		
				City Bus				
Engine, Transmission, and Fuel Weight, lb	1,500	2,200	4,300	51,000[(a)]	2,700+	2,300+	2,600+	
Engine, Transmission, and Fuel Volume, ft^3	33	46	75	130	--	46+	--	
Engine and Transmission Cost, $	Should be similar to or somewhat higher than cost of heavy-duty, commercial-vehicle diesel engine and hydrokinetic transmission			53,000[(a)]		Very high[(a)]		

(a) Designates values that are above the absolute maximum, or threshold, specification.

(b) Considered to be part of a hybrid system. Their power rating is based on the cruise power requirement which is one-half the maximum power. The weight, size, and cost of associated batteries, motors, and controls would have to be added to the values listed in this table for the converters alone.

TABLE 5. VARIATION IN ESTIMATED RANKINE-ENGINE PROPULSION-SYSTEM WEIGHT AND VOLUME FOR A SELECTION OF CHEMICAL-FUELS AND THERMAL-ENERGY STORAGE MEANS

	Chemical Fuels						Thermal
Parameter	Conventional Fuels	Propane (LPG)	Ammonia	Methane (Cryogenic)	Hydrogen (Cryogenic)	Hydrogen (Magnesium Hydride)	Lithium Fluoride
Stored Energy Specific Weight, lb/hp(t)-hr	0.15	0.22	0.51	0.27	0.44	0.74	3.30
Stored Energy Specific Volume, ft^3/hp(t)-hr	0.0029	0.0062	0.0150	0.018	0.044	0.012	0.032
Family Car							
Stored Energy, hp(t)-hr	690	690	690	690	690	690	690
Stored Energy Weight, lb	100	150	350	190	300	510	2300
Propulsion System Weight, lb	550	600	800	640	750	960	2750(a)
Stored Energy Volume, ft^3	2.0	4.3	10	12	30	8.3	22
Propulsion System Volume, ft^3	12	14	20	22	40(a)	18	32(a)
Commuter Car							
Stored-Energy, hp(t)-hr	78.4	78.4	78.4	78.4	78.4	78.4	78.4
Stored-Energy Weight, lb	12	17	40	21	34	58	260
Propulsion System Weight, lb	190	200	220	200	210	240	440
Stored-Energy Volume, ft^3	0.23	0.49	1.2	1.4	3.4	0.94	2.5
Propulsion-System Volume, ft^3	4.0	4.3	5.0	5.2	7.2	4.7	6.3
Utility Car							
Stored Energy, hp(t)-hr	64.6	64.6	64.6	64.6	64.6	64.6	64.6
Stored-Energy Weight, lb	10	14	33	17	28	48	210
Propulsion-System Weight, lb	120	120	140	130	140	160	320
Stored-Energy Volume, ft^3	0.19	0.40	0.97	1.2	2.8	0.78	2.1
Propulsion-System Volume, ft^3	2.4	2.6	3.2	3.4	5.0	3.0	4.3
Delivery Van							
Stored-Energy, hp(t)-hr	315	315	315	315	315	315	315
Stored-Energy Weight, lb	47	69	160	85	140	230	1000
Propulsion-System Weight, lb	500	520	610	540	590	680	1450
Stored-Energy Volume, ft^3	0.91	2.0	4.7	5.7	14	3.8	10
Propulsion-System Volume, ft^3	11	12	15	16	24	14	20
City Taxi							
Stored Energy, hp(t)-hr	553	553	553	553	553	553	553
Stored-Energy Weight, lb	83	120	280	150	240	410	1800
Propulsion-System Weight, lb	340	380	540	410	500	670	2100(a)
Stored-Energy Volume, ft^3	1.6	3.4	8.3	10	24	6.6	18
Propulsion-System Volume, ft^3	7.2	9	14	16	30(a)	12	24
City Bus							
Stored Energy, hp(t)-hr	2120	2120	2120	2120	2120	2120	2120
Stored-Energy Weight, lb	320	470	1100	570	930	1600	7000
Propulsion System Weight, lb	2200	2350	2980	2450	2810	3480	8880(a)
Stored-Energy Volume, ft^3	6.2	13.0	32	38	93	25	68
Propulsion System Volume, ft^3	46	53	72	78	130	65	110

(a) Designates values that are above the absolute maximum, or threshold, specification.

TABLE 6. ESTIMATED FLYWHEEL AND ASSOCIATED TRANSMISSION CHARACTERISTICS FOR FLYWHEEL-ENGINE HYBRID VEHICLE PROPULSION SYSTEMS

Parameter	Vehicle Application	
	Family Car	City Bus
Assumed Speed, mph	60	30
Assumed Effective Mass, lb	4,400	33,000
Vehicle Kinetic Energy, hp(s)-hr	0.27	0.50
Maximum Delivered power, hp(s)	94	180
Assumed Transmission and Drive Efficiency, percent	80	80
Flywheel Energy, hp(s)-hr	0.34	0.62
Transmission Power, hp(s)	117	225
Flywheel Specific Weight, lb/hp(s)-hr	100	100
Flywheel Weight, lb	34	62
Transmission Specific Weight, lb/hp(s)	2.0	3.0
Transmission Weight, lb	230	680
Flywheel and Transmission Weight, lb	260	740
Flywheel Specific Volume, ft^3/hp(s)-hr	0.25	0.25
Flywheel Volume, ft^3	0.085	0.16
Transmission Specific Volume, ft^3/hp(s)	0.02	0.03
Transmission Volume, ft^3	2.3	6.75
Flywheel and Transmission Volume, ft^3	2.40	6.9
Flywheel Specific Cost, $/hp(s)-hr	100	100
Flywheel Cost, $	34	62
Transmission Specific Cost, $/hp(s)	3.0	14.0
Transmission Cost $	350	3,200
Flywheel and Transmission Cost, $	380	3,300

duty, long lived, and maintenance free and because it is a lower production-quantity item. For these reasons, the specific weights and volumes of the Rankine and Stirling engines that were estimated for the bus application are twice as large as those listed in Table 2 for automobile-type applications. Since gas turbines and direct converters are inherently more durable, maintenance-free devices, their specific weights and volumes were increased by only 33 percent for the bus application. It is believed that the cost for any of the heat engines plus required transmissions for the bus application would be similar to or somewhat higher than the cost of commercial-vehicle diesel engines plus hydrokinetic transmissions.

The other five applications of concern currently use, and it is believed will continue to use, automobile-type propulsion systems. Thus, it was not necessary to alter the characteristic values listed in Table 2 for any of these applications.

Average engine efficiency over the various duty cycles was assumed to be approximately 70 percent of projected peak engine efficiency. Since the direct converters will operate under more nearly study-state conditions and nearer to their design point, their average efficiency was assumed to be approximately 90 percent of projected peak efficiency.

The heat-engine propulsion-system weights and volumes listed in Table 3 include those of the engine, transmission, and fuel and fuel-storage provisions only. The listed initial costs are those of the engine and transmission only. These represent the major portions of the propulsion system, and the addition of exhaust and driveline, other than transmission, components would neither significantly increase estimated system weights, volumes, and costs nor, more importantly, alter the conclusions resulting from these estimates.

As stated above, because of the basic operating nature of the direct converters, they would probably be used as part of a hybrid system along with suitable batteries, motors, and controls. The weights and volumes listed for the converters in Table 3 include those of the converters and fuel and fuel-storage provisions only. Thus, the weights and volumes of associated batteries, motors, and controls must be added to those listed for the direct converters before a direct comparison can be made between the weights and volumes of the direct-converter systems and those of the engine and flywheel systems.

The values listed in Table 4 represent an optimistic estimate of the best physical characteristics that could be expected to be achieved for each of the alternative systems 10 to 15 years from now, assuming that major research and development efforts were expended on them. Gas-turbine characteristics are not listed for the utility-car application because it is not believed that a competitive small gas turbine [i.e., less than 30 hp(s)] could be developed. The estimated heat-engine characteristics calculated for the low-energy and -power, commuter-car and utility-car applications are believed to be somewhat overly optimistic, but are adequate for the purposes of this study.

In both Tables 3 and 4, fuel-storage characteristics were calculated assuming that one of the conventional liquid fuels plus a suitable sheet-metal tank would be used for energy storage. To determine the effect that the use of alternative chemical-fuel and thermal-energy storage means would have on propulsion-system weight and volume, the variation in Rankine (steam)-engine propulsion-system characteristics was calculated for a selection of chemical-fuel and thermal-energy storage means, including suitable container provisions. The results of these calculations are given in Table 5.

The assumed effective masses listed in Table 6 were arrived at by increasing the loaded weights of the family car and city bus 10 percent to account for the effective inertia of rotating parts. The transmission characteristics listed apply to hydrostatic transmissions which, it is believed, would be most appropriate for use with a flywheel-engine hybrid system.

The maximum cruising velocities listed in Table 1 are greater than those to which the acceleration specifications are set for the family-car and city-bus applications. Thus, if it is assumed that the engines in the hybrid systems are rated to supply maximum cruising power, the engines would be capable of supplying the energy required to overcome all rolling and aerodynamic vehicle loadings during the acceleration period and would provide some of the energy required for accelerating the vehicles. By making the energy-storage capacity of the flywheels equivalent to the kinetic energy of the vehicles at the speeds to which the acceleration specifications apply, a suitable reserve of energy is provided. This energy reserve could be used to accelerate the vehicles to speeds somewhat higher than that to which the acceleration specifications apply.

SYSTEM EVALUATION

As previously discussed, the alternative propulsion systems and components were evaluated as to their suitability for urban-vehicle application by comparing their projected "1980" characteristics with the hypothesized threshold specifications for various urban-vehicle missions listed in Table 1. In this manner, the obviously unacceptable propulsion systems were identified and the relative attractiveness of the systems "above threshold" was determined.

Through such comparisons, magnetohydrodynamics and all nuclear devices were found to be unsuitable for urban-vehicle application early in the evaluation process and, thus, were not subjected to more detailed analysis. The other alternative systems included in this study, however, were evaluated in more detial and, as discussed in the preceding section, their estimated weights, volumes, and initial costs are listed in Tables 3 through 6.

All of the characteristic values listed in Tables 4 through 6 that do not meet the threshold specifications listed in Table 1 are designated by (a). Systems which are not expected to meet the threshold specification with respect to one or more characteristics for any given urban-vehicle application are concluded to be unsuitable for the particular application in question.

Study of Table 4 shows that of the alternative systems and devices studied, only the three heat engines are expected to exceed all of the threshold specifications. Magnetohydrodynamic and nuclear devices were excluded earlier. Flywheel energy storage by itself is unacceptable from a weight and cost standpoint for all six urban-vehicle applications, and its estimated volume meets the threshold specifications for only three of the six applications. Finally, the projected specific cost [i.e., \$100/hp(e)] for the direct converters results in converter costs higher than the threshold specification for all six applications.

The system characteristics estimated for gas turbines and Rankine (steam) engines are well above threshold performance, and it is concluded that these systems are very attractive alternative propulsion systems for general urban-vehicle application. This is particularly true in light of their low-pollutant-emission characteristics which are discussed in the following section on the state-of-the-art of these engines.

While the estimated system characteristics for Stirling engines also exceed threshold specifications in all cases, they are less attractive than those estimated for gas turbines and Rankine engines. Thus, it is concluded that Stirling engines are not as attractive for general urban-vehicle applications as Rankine engines and gas turbines. The principal exception to this may be the bus application, where the superior efficiency of the Stirling engine and its potential for high durability may make it a preferred alternative.

While the costs projected for the direct converters could not meet the threshold specifications, projected costs are subject to the greatest variation, as noted in Table 2, so it is prudent to also consider the relative attractiveness of the direct converters and the heat engines on a strictly weight, volume, and efficiency basis. Again referring to Table 4, it can be seen that even without adding the additional weight required for batteries, motors, and controls, the weight listed for the direct converters is much higher than that for the gas-turbine and Rankine-engine systems for the family-car, delivery-van, and taxi applications. The weight difference is not so great for the other applications, but in no case is the estimated weight of the direct-converter systems, less batteries, motors and controls, lower than that estimated for the gas-turbine and Rankine-engine systems.

The other major consideration, of course, is efficiency. In Table 3 and the tables in Appendix B, estimated average efficiencies for the direct converters are one-third to two-thirds of those for the heat engines. This comparison is even less favorable for the direct converters if estimated peak efficiences are compared. The direct converters' low efficiency would result in increased fuel consumption which would in turn result in increased operating cost and heat release. Exhaust emissions may or may not be increased by this higher fuel consumption, as the direct converters' use in hybrid systems would mean that their burners would operate at more nearly steady-state conditions and thus probably emit less pollutant per pound of fuel burned.

Because of their very high projected cost and particularly because of their low projected efficiency and high weight, direct thermal-to-electric energy converters are not considered to be attractive alternatives for urban-vehicle propulsion systems.

The data listed in Table 5 show that the use of thermal-energy storage with the Rankine engine would result in a system that would not meet threshold specifications for any except the lowest energy vehicle applications. The lithium fluoride system is heavier and larger than specified for the family-car application, heavier than specified for the taxi and bus applications, and just under the maximum allowable weight for the delivery-van application. While somewhat smaller and lighter thermal-energy storage systems could be used with Stirling engines, they would still be unattractive for these high-energy applications. Thus, thermal-energy storage is suitable only for the very low energy applications, and for such applications, it would encounter very strong competition from battery systems.

Assuming a 25 percent average engine efficiency over the applicable driving cycle, which is the average efficiency estimated for the Stirling engine in the commuter-car application, a lithium fluoride-energy storage device might have a specific weight of 13 lb/hp(s)-hr [3.30 lb/hp(t)-hr ÷ 0.25]. This specific weight is higher than that presently being projected for some advanced electrochemical batteries [i.e., 6 to 8 lb/hp(e)-hr for sodium-sulfur batteries]. Thus, it is questionable that thermal-energy storage would be found to be the best alternative even for low-energy, urban-vehicle applications.

Of the alternative chemical-fuel-energy storage means listed in Table 5, only the cryogenically stored hydrogen system could not meet threshold specifications for all of the vehicle applications. Its volume was found to be excessive for both the family-car and city-taxi applications. While the use of alternative fuels generally resulted in systems that could meet threshold specifications, study of Table 5 shows that they are considerably less attractive from a weight and volume standpoint than the more conventional fuels (i.e., gasoline, JP-4, kerosine, and diesel fuel). Only the liquid petroleum gases (LPG) would result in energy-storage weights and volumes relatively close to those of the conventional fuels. The alcohols would have a size and weight disadvantage similar to that of LPG; however, they are more costly and are not known to have advantageous emission characteristics as has been cited for LPG in piston engines.

The energy specification for the commuter car and utility car are small so the effect of heavier and larger energy-storage provisions is considerably less pronounced on their total-system characteristics than for the other applications. Their short range (50 miles), however, was set primarily out of consideration of a battery system which could be recharged every night. If a chemical-fuel-energy storage means requiring a stop at a service station for refueling were used, however, it would probably be desirable to increase this range, which would in turn increase the weight and size disadvantage of using an alternative fuel.

Because of the above-discussed weight and size requirements of the fuel plus appropriate container, it is concluded that the unconventional chemical fuels investitated, with the possible exception of the LPG fuels, are not attractive for general use in urban vehicles. The principal exception to this would be the very low energy vehicle applications. To overcome their weight and size handicap, unconventional fuels such as ammonia, methane, and hydrogen, would have to possess emission characteristics considerably better than those of conventional fuels before they could be considered as legitimate contenders for urban-vehicle application. When considering external combustor systems, it is believed unlikely that these latter unconventional fuels will possess sufficiently improved emission characteristics, particularly with respect to NO_x, to compensate for their weight and size handicap.

While the flywheel has poor energy-storage characteristics [i.e., 100 lb/hp(s)-hr], it is limited with respect to how fast stored energy can be removed, or to power, only by the size of the associated shafting and transmission. Thus, in a flywheel-engine hybrid system, a chemical-fueled engine would supply the gross amount of energy required for a given vehicle application, and only that energy required for vehicle acceleration need be stored in the flywheel. The flywheel could then be "recharged" between vehicle accelerations.

Study of Table 6 shows that surprisingly small flywheels would be required for use with flywheel-engine hybrid systems. The biggest change from current systems would be with respect to the transmission requirements, particularly for the family-car application. Even though a lower power engine could be used, the addition of the flywheel and particularly the need for a larger and more costly transmission make questionable the overall desirability of a flywheel-engine hybrid system for the family car application. For the bus application, where frequent stopping and starting is encountered and where heavy-duty and costly engines and transmissions are already used, however, a flywheel-engine hybrid system may be desirable.

The flywheel-engine hybrid system would be a competitor to a battery-engine hybrid system; thus, the two systems must be studied and compared in detail before it can be determined which is the more attractive of the two.

PRESENT AND PROJECTED STATE OF THE ART FOR ALTERNATIVE SYSTEMS

- CHEMICAL FUELS AND EXTERNAL COMBUSTORS
  - CHEMICAL FUELS
  - EXTERNAL COMBUSTORS
- HEAT ENGINES
  - BRAYTON CYCLE (GAS TURBINE) ENGINES
  - RANKINE CYCLE (STEAM) ENGINES
  - STIRLING CYCLE (HOT AIR) ENGINES
- DIRECT THERMAL-TO-ELECTRIC ENERGY CONVERTERS
  - MAGNETOHYDRODYNAMICS
  - THERMOELECTRIC CONVERTERS
  - THERMIONIC CONVERTERS
  - THERMOPHOTOVOLTAIC CONVERTERS
- OTHER ENERGY STORAGE AND CONVERSION DEVICES
  - THERMAL ENERGY STORAGE
  - MECHANICAL ENERGY STORAGE
  - VEHICLE TRANSMISSIONS
  - NUCLEAR DEVICES

PRESENT AND PROJECTED STATE OF THE ART FOR ALTERNATIVE SYSTEMS

This section of the report contains detailed discussions of the present and projected physical, performance, and cost characteristics for all of the alternative systems and devices investigated during the program. The projected characteristics are for a period 10 to 15 years in the future (i.e., "1980" characteristics). The discussions of the individual devices and systems contained in this section are organized as illustrated on the preceding divider sheet.

CHEMICAL FUELS AND EXTERNAL COMBUSTORS

CHEMICAL FUELS

Fuels Investigated

For the purposes of this study, twelve specific chemical fuels were selected as the most likely candidates for urban-vehicle application and were studied to determine their physical and cost characteristics. The fuels selected fall into two general categories: those that are liquids at ambient temperatures and pressures, and those that are gases. The liquid fuels, i.e., gasoline, JP-4 (JP-5 and CITE also), kerosine, No. 2 diesel oil, methanol, and ethanol, represent a class of petroleum products which have been found to be particularly appropriate as engine or prime mover fuels. The gaseous fuels include propane and butane, commonly referred to as the liquefied petroleum gases (LPG), methane and ethane, the principle constituents of most natural gases, and hydrogen and ammonia.

Current Availability and Application of Fuels

Gasoline, JP-4, kerosine, and diesel oil are readily available in either large, medium, or small quantities, through bulk distributors and retail outlets such as service stations. All are used as prime mover fuels, and kerosine and diesel oil are also used extensively in small-to-medium size industrial, commercial, and residential heating and hot-water systems.

Ethanol and methanol are currently available in 55-gallon drums and also in smaller quantities. Their chief uses are as industrial solvents, in the preparation of cooling-system antifreeze solutions, and in laboratory experiments.

Ammonia is becoming more and more readily available in bulk quantities for agricultural use as a fertilizer. The military is also doing some experimenting with ammonia as an engine fuel.

Butane and propane are available from bulk distributors in both large and small quantities, generally for residential and mobile (trailer) heating and cooling. A small amount of propane is also used as a vehicle engine fuel, principally for inside operating lift trucks and similar vehicles, and for the tank trucks used to transport the propane.

Methane is available as the major (generally over 90 percent) constituent of natural gas, which is supplied by pipeline from gas fields in Texas, Oklahoma, and other areas to consumers throughout most populated sections of the U. S. The main uses of natural gas are industrial, commercial, and residential heating and air conditioning, manufacturing operations such as heat treating, drying, and mineral processing, and cooking. Methane is also available in high-pressure cylinders for laboratory use. Ethane is not readily available except as an ingredient in some natural gases.

Hydrogen is available in high-pressure cylinders for laboratory use, in bulk at atmospheric pressure and temperature for some industrial processes such as in petroleum refining, and in liquid form for cryogenic purposes.

Current and Projected State of the Art

Physical and Cost Characteristics – Fuels

Table 7 contains a listing of some pertinent physical characteristics and current cost information for the fuels investigated. The physical data were obtained from various reference books and other sources.(1-5)* The lower heating value represents the total energy in the fuel that could be converted to useful work in an ideal combustion system. The stoichiometric air-fuel ratio represents the minimum amount of air (on a weight basis) required for complete combustion. The flame temperatures are theoretically calculated assuming perfect mixing and stoichiometric conditions. The boiling point at 1 atmosphere indicates the temperature above which the fuel will no longer remain in liquid form at atmospheric pressure.

The current cost data were obtained from a number of diverse sources.(6-10) The costs of the engine fuels (gasoline, kerosine, and diesel oil) are based on approximate local (Columbus, Ohio) service station prices, exclusive of road taxes; JP-4, being similar to kerosine, is estimated to cost the same; the costs of ethanol and methanol are based on fairly large quantity commercial prices; the cost of ammonia is based on the selling price of anhydrous ammonia for fertilizer use in quantities normally associated with farm use; the costs of propane and butane are based on approximate local prices for trailer and home heating use; and the costs of methane and hydrogen are based on rough estimates of the cost of providing these gases at 3000 psig pressure or at cryogenic conditions. At atmospheric pressure, methane costs about 60 cents per 1000 cu ft, or 1.4 cents per pound, and hydrogen might cost about 40 cents per 1000 cu ft or 8 cents per pound. No cost data could be located for ethane.

Because of the great differences in the present production quantities, sources of supply, types of distribution systems and general end uses for each of the fuels, current cost data are being listed here only as a limiting case. That is, the listed costs represent a maximum potential cost for the fuels in question. If produced and distributed in the quantities and manner that the current engine fuels are, the cost of many of the fuels

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*Superscript numbers refer to references listed in Appendix C.

TABLE 7. PERTINENT PHYSICAL CHARACTERISTICS AND CURRENT COSTS OF SELECTED FUELS

Fuel	Lower Heating Value, Btu/lb	Density lb/cu ft	Flame Temperature, F	Stoichiometric Air-Fuel Ratio	Boiling Point at 1 Atm, F	Current Cost, ¢/lb
Gasoline	19,100	46.3	4190	14.8	145/340	3.5
JP-4, JP-5, CITE	18,800	48.5	--(c)	14.5	150/550	3.0
Kerosine	19,200	51.0	--(c)	15.0	370/500	3.0
Diesel No. 2	19,000	52.2	3850	15.0	400/590	2.7
Ethanol	11,700	49.1	--(c)	9.0	172	13.9
Methanol	8,700	49.7	--(c)	6.4	149	9.5
Ammonia(a)	8,000	40.2	3092	6.1	-27	4.2
Butane(a)	19,700	35.6	3583	12.2	31	6.0
Propane(a)	20,000	36.2	3573	5.5	-44	4.0
Ethane(b)	20,600	16.1	3540	12.4	-128	--(c)
Ethane(d)	20,600	34.0	3540	12.4	-128	--(c)
Methane(b)	21,600	8.5	3484	12.6	-259	2.1
Methane(d)	21,600	25.9	3484	12.6	-259	2.4
Hydrogen(b)	51,620	1.1	4010	34.3	-423	14.0
Hydrogen(d)	51,620	4.47	4010	34.3	-423	20.0

(a) In liquid state at 80 F.
(b) In gaseous state at 80 F and 3000 psig.
(c) No data.
(d) In liquid state at atmospheric pressure and cryogenic temperature.

would be less than the current values listed. This lack of sound, comparable cost information also applies to the various containers for these fuels. Appropriate containers can range from nonpressurized tanks to low-or high-pressure, pressure vessels requiring suitable safety provisions for automotive application, to cryogenic storage systems with or without an on board cooling means. Current costs for such containers are not directly comparable because of the great differences in utilization and production quantities.

Figure 2 shows the relationship between vapor pressure and temperature for the six gaseous fuels. The curve for each fuel essentially represents the line of differentiation between the gas and the liquid state; if the existing temperature and pressure define a point to the left of the line the fuel will be liquid, and if they define a point to the right of the line it will be a gas. It will be noted that all of these fuels are gases at

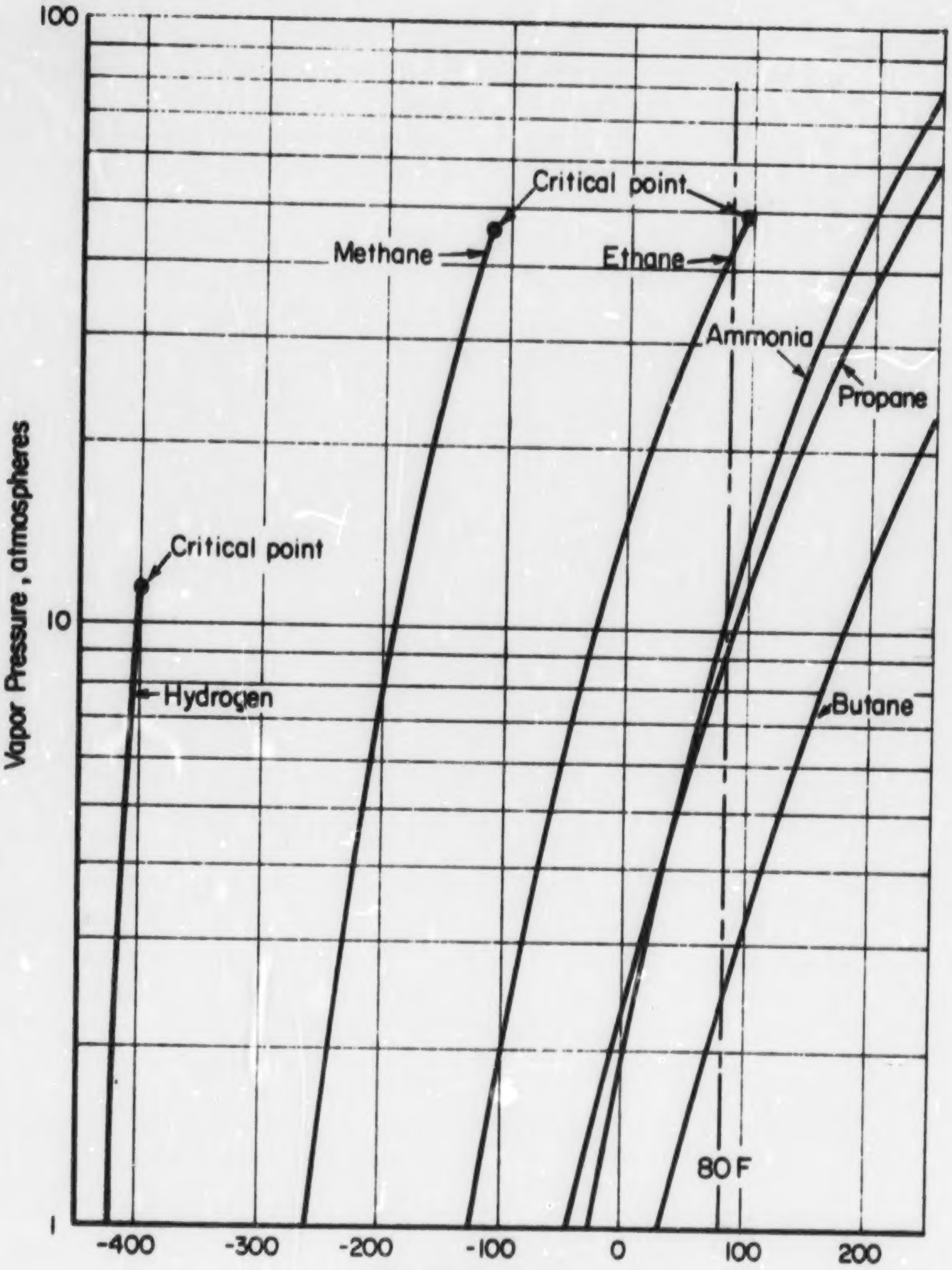

FIGURE 2. VAPOR PRESSURE VERSUS TEMPERATURE FOR LOW-BOILING-POINT FUELS(3)

atmospheric pressure and at an ambient temperature of 80 F. On the other hand, all of these fuels are sometimes shipped, stored, and used in the liquid form, and in order to do this, either the pressure must be raised, the temperature lowered, or a combination of both.

Referring to the curves of Figure 2, it is evident that raising the pressure to 10 to 15 atmospheres would be sufficient to maintain butane, propane, and ammonia in the liquid state; and this is actually standard procedure in the handling of these fuels. The normal pressure rating on LPG tanks is about 200 psi gage. The great advantage is, of course, reduced bulk for a given amount of fuel being handled. Ethane requires a pressure of about 43 atmospheres to remain in the liquid state at 80 F. Methane and hydrogen cannot be maintained in the liquid state at 80 F under any conditions. The general practice with these latter two fuels, when it is desired to handle them as liquids, is to reduce their temperature at atmospheric pressure. For instance, methane will remain liquid below -260 F at atmospheric pressure and hydrogen will remain liquid below -423 F at atmospheric pressure.

Physical Characteristics – Fuel Plus Container

As discussed above, appropriate containers for the various fuels can vary widely as to their weight, size, and complexity. Therefore, when comparing alternative fuels for vehicular application, it is not sufficient to consider the characteristics of the fuels alone. Rather, it is necessary to compare the characteristics of the fuels plus appropriate container provisions.

Table 8 contains specific weight, volume, and current cost data for the twelve fuels of interest expressed in units that are more suited to the purposes of this study. These data were derived from the physical and cost data listed in Table 7. However, Table 8 also contains data on the specific weight and volume of the fuels plus compatible container provisions.

The fuel plus container weights were estimated as follows. First, a container volume of 2 cu ft, or 15 gallons, was assumed. Then, for the normally liquid fuels, a simple rectangular steel tank made of 24-gauge material was assumed. For the liquefied petroleum gases and ammonia, a conventional commercially available LPG tank weight was used, with some allowance being made for adapting the tank for vehicular application. For the natural gases and hydrogen, catalog data were used for both the high-pressure container systems and the cryogenic container systems. A spherical fiber-glass tank rated at 3000 psi was used for the pressurized fuel system, and a spherical vacuum-insulated cryogenic tank with an estimated boil-off rate of 2 to 5 percent per day was used for the cryogenic fuel system. The weights determined for these 2-cu ft containers were then added to the weights of the contained fuel to determine the specific weights on a fuel-plus-container basis.

The fuel-system volumes were also estimated on the basis of a 2-cu ft container capacity. For the normally liquid fuels it was assumed that the tanks could be readily shaped to fit any reasonable space on the vehicle; hence the total fuel system volume was estimated to be only a small percentage (approximately 8 percent) greater than the fuel volume alone to allow for the container material and rounded corners. For the liquefied petroleum gases and ammonia fuels, actual volumes of commercial LPG tanks were

TABLE 8. SPECIFIC WEIGHT, VOLUME, AND CURRENT COST OF SELECTED FUELS AND FUEL PLUS CONTAINER

	Fuel Alone			Fuel and Container	
Fuel	Specific Weight, lb/hp(t)-hr	Specific Volume, ft^3/hp(t)-hr	Specific Current Cost, \$/hp(t)-hr	Specific Weight, lb/hp(t)-hr	Specific Volume, ft^3/hp(t)-hr
Gasoline	0.133	0.00290	0.0047	0.15	0.0031
JP-4, JP-5, CITE	0.136	0.00280	0.0041	0.15	0.0030
Kerosine	0.133	0.00261	0.0040	0.14	0.0028
Diesel No. 2	0.134	0.00256	0.0036	0.15	0.0028
Ethanol	0.218	0.00444	0.0300	0.24	0.0048
Methanol	0.293	0.00590	0.0280	0.32	0.0064
Ammonia(a)	0.319	0.00805	0.0130	0.51	0.0150
Butane(a)	0.129	0.00366	0.0078	0.21	0.0064
Propane(a)	0.127	0.00351	0.0051	0.22	0.0062
Ethane(b)	0.124	0.00770	--(c)	0.37	0.0170
Ethane(d)	0.124	0.00365	--(c)	0.25	0.0140
Methane(b)	0.118	0.01390	0.0025	0.56	0.0310
Methane(d)	0.118	0.00456	0.0028	0.27	0.0180
Hydrogen(b)	0.049	0.0455	0.0070	1.49	0.1000
Hydrogen(d)	0.049	0.0114	0.0100	0.44	0.0440

(a) In liquid state at 80 F.
(b) In gaseous state at 80 F and 3000 psig.
(c) No data.
(d) In liquid state at atmospheric pressure and cryogenic temperature.

used, again with some allowance for adaptation to vehicular use. For the natural gases and hydrogen, although the tanks were assumed to be spherical to minimize weight, the space required on the vehicle was assumed to be only slightly less than the volume of the cube with sides equal to the diameter of the sphere.

Because of practical considerations and insulation requirements, selection of container volumes greater than 2 cu ft would likely result in somewhat lower specific weights and volumes, and volumes lower than 2 cu ft in somewhat higher specific weights and volumes. Because of the nature of this study and the cursory nature of the container information available, however, such variation is not considered significant to the conclusion of the study.

The data in Table 8 illustrate the superiority of the conventional fuels (gasoline, JP-4, kerosine, and diesel oil) on a weight and volume basis. It should be recognized that the low cost of these four fuels derive to some extent from the exceedingly great demand for their use in vehicular, aircraft, and stationary powerplants. However, this demand is in itself predicated on their eminent suitability for the purpose.

In connection with the cryogenic storage of natural gases and hydrogen, there is the problem of boil-off rate which can vary from 2 to 20 percent per day, depending on the quality of insulation and construction of the tank. A 20 percent boil-off rate would probably be unacceptable because of the poor fuel mileage that would result. A 2 percent boil-off rate might be extremely expensive to achieve in practice.

The selection of 3, 000 psi for the pressure at which the natural gases and hydrogen would be stored in the gaseous state is arbitrary. A higher pressure would naturally result in greater energy storage per unit volume, both for the fuel alone and for the fuel plus tank and auxiliaries. It would however, result in a similar energy storage per unit weight of fuel plus tank.

There appears to be another alternative in the means of supplying and storing hydrogen for vehicular use. This alternative is the use of magnesium hydride as an adsorber-desorber for hydrogen gas at atmospheric pressure, a technique under study at the Brookhaven National Laboratory.(11) Briefly, a system based on this technique would operate on a cycle involving the generation of hydrogen gas for use, by heating the magnesium hydride and regenerating the magnesium hydride by passing hydrogen gas through it while extracting heat.

The performance capabilities of a magnesium hydride fuel system sized for a 45-pound hydrogen capacity (approximately 920 hp(t)-hr) were estimated using the following assumptions: hydrogen available for regeneration at 40 cents per 1000 cu ft, including the costs of the regeneration process; 547 pounds of magnesium-nickel alloy (5 percent nickel) at 42 cents per pound, including the cost of pulverizing to an average particle size of 150 microns; and a metal container and heat exchanger (for generation of the gas) weighing a total of 100 pounds and costing $1. 00 per pound. The resulting system performance was as follows: 0. 74 lb/hp(t)-hr for fuel and system, 0. 012 ft^3/hp(t)-hr for fuel and system, and 0. 35 $/hp(t)-hr for the fuel system alone. At 40 cents per 1000 cu ft, the cost of the fuel only is 8 cents per pound or 0. 004 $/hp(t)-hr.

Comparing the above data with that listed in Table 8 indicates that the magnesium hydride storage method would result in a significantly lower specific volume but a somewhat higher specific weight than with the cryogenic method. However, consideration of a fuel system of the magnesium hydride type must take into account the fact that the development is still in an embryonic stage and that several problems remain to be solved before a more complete assessment of its potential can be made.

EXTERNAL COMBUSTORS

General Description

For this study an external combustor can be defined as a device for burning a fuel and air (or oxidizer) mixture for the purpose of producing heat which can be converted to useful work. The combustion takes place outside of (or external to) the device which converts the heat to work. External combustors are used in connection with Rankine-cycle (steam) engines, Stirling-cycle engines, and direct thermal-to-electric energy converters. Although in the true theoretical sense, gas-turbine combustors are not "external" combustors, the fact that they operate under steady-state conditions and are a separate, distinguishable component results in their having characteristics very similar to those of external combustors.

In addition to their use with heat engines and direct converters, there are many examples of external combustors which produce heat for space heating, for chemical processes, for water heating, and for cooking. In many cases these same combustors with modifications could be adapted to produce heat for conversion to shaft or electrical power. However, for the most part, combustors for stationary or fixed installations are too bulky, too heavy, and too limited in control range and response to be adaptable to a vehicular application.

For maximum compactness and efficiency, an external combustor must be designed integrally with the energy-conversion device which produces the useful work output. However, all combustors will have certain basic configuration features in common, including: a fuel system, an ignition system, an air supply system, and a combustion system. The fuel system may consist of an injector or other device (e. g., ultrasonic) for atomizing the fuel and introducing it into the combustion-air stream, a pump for transporting the fuel from the tank to the injector, and a fuel-rate control. The ignition system might consist of a spark plug or a glow plug with an appropriate power supply and control switch. Either ignition source could be continuously energized or energized only during the starting sequence. The air supply system might include a blower, flow control dampers, and a means for directing and distributing the air to the proper chambers and passages within the combustor. The combustion system might consist of a mixing zone where the fuel and air are mixed before ignition, a primary combustion zone where most of the combustion takes place, a secondary combustion zone where combustion is completed with added combustion air, and a dilution zone where more air is added to the combustion gases to produce the final desired outlet gas temperature.

Current and Projected State of the Art

As mentioned earlier, an external combustor should be designed integrally with the energy-conversion prime mover to insure maximum compactness and efficiency and minimum cost. Consequently, it is impractical to compare on an absolute basis the many different external combustor configurations that would be needed by the different power sources and fuel types being considered in this study. However, an appreciation of the approximate relationship between combustor size and energy output can be obtained by considering a "general case" combustor.

Combustor Performance

A commonly used measure of the performance for combustors in the heat-release rate per cubic foot of combustion-chamber volume per atmosphere of pressure. Heat-release rates of combustors are a function of the rate at which air can be passed through them at a satisfactory pressure drop. In high-pressure-drop combustors, such as aircraft gas turbines, heat-release rates as high as 15 x 10^6 Btu/hr-ft^3-atm are used. For industrial process heaters, on the other hand, heat-release rates as low as 0.5 x 10^6 Btu/hr-ft^3-atm are used.

A combustor for an external combustion vehicular powerplant would be designed for lower air-flow capacity and a lower overall pressure drop than a gas-turbine combustor, to reduce noise and to minimize blower requirements. It would also be designed to operate at atmospheric pressure. Therefore, a heat-release rate of 1 x 10^6 Btu/hr-ft^3 (390 hp(t)/ft^3 can be assumed as a reasonable combustion-chamber volume for comparison pruposes. Assuming that the volume required for the air blower and ducting and the other auxiliaries is equivalent to the combustion-chamber volume results in a likely specific-heat-release rate of approximately 200 hp(t)/ft^3 for an applicable combustor.

A combustor will produce about the same amount of power regardless of the type of fuel used. The reason for this is that, as mentioned, combustors are limited not by the amount of fuel that can be pumped through the system but by the amount of air, and the potential heat released per pound of air at stoichiometric combustion conditions is very nearly a constant for the fuels of interest (about 1300 Btu/lb air). There are two exceptions to this, hydrogen and ammonia. With hydrogen the potential heat release is about 1500 Btu/lb air, thus, the same-size external combustor might be expected to produce about 15 percent more power when burning hydrogen. Ammonia, though right in line with the other fuels on heat released per pound of air, would produce less power than the other fuels in the same size combustor because of ignition and burning-rate difficulties peculiar to the ammonia fuel.

Performance of Hydrogen and Ammonia in Internal-Combustion Engines

While the scope of this program excludes internal-combustion (IC) engines, it is desirable to comment on the potential performance of two fuels, hydrogen and ammonia, in IC engines because they are not normally considered as fuels for that type of engine.

Assuming that the two fuels would be carbureted or injected into the engine in the vaporized liquid form and the operating air-fuel ratio would be in the same relationship to the stoichiometric ratio as for gasoline, ammonia fuel should theoretically produce nearly the same power from the same-size, naturally aspirated engine as gasoline fuel, and hydrogen should produce about 15 percent more power (because of its higher potential heat release per pound of air). If, on the other hand, the ammonia and hydrogen fuels were carbureted into the engine in the gaseous form, which might be a logical approach to promote good mixing of the fuel and air and to minimize fuel system cost and complexity, theory indicates that the ammonia fuel would produce about 22 percent less power than gasoline fuel in the same naturally aspirated engine and hydrogen fuel would produce about 19 percent less power. In the gaseous form the fuel displaces some of the air that would normally be inducted into the engine, thereby reducing the total weight of fuel and air in the engine per cycle. As a comparison, the same engine using natural gas as the fuel (carbureted) would theoretically produce about 12 percent less power than gasoline fuel.

In actual experimental operation of internal-combustion engines on ammonia fuel(12-14), about 70 percent of the rated power of the engine on gasoline fuel was produced when gaseous ammonia was carbureted into the engine, and about 90 percent of the rated power was achieved when liquid ammonia was injected into the engine induction air. It was surmised that the full theoretical performance of ammonia as an engine fuel was not achieved because of ignition difficulties and slow burning of the ammonia-air mixture. These problems could probably be somewhat lessened with development.

Emission Characteristics of External Combustors

The air pollutants resulting from the combustion of fuel oil and natural gas in various types of external combustors has received a good deal of attention. The emphasis has been primarily on the control of smoke and oxides of sulfur, which are the principal problems associated with stationary combustion sources. It is notable that similar combustion sources using the same fuel type can produce significantly different contaminant-emission levels. This fact points up the importance of burner design, adjustment, and maintenance in controlling exhaust emissions.

Table 9 is a compilation of emission data for natural-gas- and oil-fired combustion equipment of various types in residential, commercial, and industrial applications. In general, these data show that natural-gas combustion results in lower exhaust-pollutant-emission levels than fuel oil and that pollutant emissions from stationary external-combustion sources are very low. Table 10 is a compilation of emission data for external combustors associated with Stirling and Rankine (steam) engines. Considerable variation exists in these data also.

As a comparison to the emission data in Tables 9 and 10, a standard gasoline engine without exhaust controls might produce the following emission per 1 million Btu of fuel consumed: 2.7 lb unburned hydrocarbons (HC), 0.79 lb oxides of nitrogen (NO_x), 0.04 lb oxides of sulfur (SO_x), 0.012 lb smoke, and 0.09 lb aldehydes(22). Another reference(25) gives the emissions from a typical conventional gasoline engine without exhaust controlls as: 900 ppm HC, 3.5 percent CO, and 1500 ppm NO_x.

Study of the data in question indicates that on a mass-emission-rate basis (mass of emissions per unit energy of fuel burned), external combustors generally emit less than 4 percent of the CO and HC that uncontrolled gasoline engines do and vary widely in their emission of NO_x. The NO_x emissions indicated in Tables 9 and 10 vary from roughly 5 percent to equivalent amounts of that emitted by uncontrolled gasoline engines.

Studies have shown that the emission characteristics of most external combustors are highly dependent on air/fuel ratio and inlet air temperature. General observation of many combustors indicates that increasing the air/fuel ratio (increasing excess air) beyond the stoichiometric ratio will increase No_x and SO_x and will decrease HC, CO, and smoke. These affects could be expected on the basis that more air means more oxygen, both to participate in the formation of NO_x and SO_x and to promote more

TABLE 9. EMISSION DATA FOR NATURAL-GAS- AND OIL-FIRED COMBUSTION EQUIPMENT

Fuel	Type of Equipment	Reference	Emission Levels, lb per 1,000,000 Btu					
			CO	HC	NO_x	SO_x	Smoke	Aldehydes
Gas	Industrial and commercial burners, 100-4000 ft^3/hr (avg of 12)	15	(<.01)[(a)]	--	.10	--	.02	.01
Oil	Industrial and commercial burners, 5-15 gph (avg of 6)	15	(<.01)[(a)]	--	.15	.42	.10	.02
Oil	Test furnace, 1 gph, 75% excess air	16	.054	.002	.065	.10	--	--
Oil	General stationary combustion sources	17	--	--	.71	.16	.01	--
Gas	General stationary combustion sources	17	--	--	.45	--	--	--
Oil	Power-plant boilers	18	--	--	.62	1.4	.16	--
Gas	Power-plant boilers	18	--	--	.22	Negl.	.02	--
Gas	Domestic appliances	19	--	--	.08	--	--	.01
Oil	Furnaces, boilers, and power plants	20	.027	.004	.49	--	.11	.01
Oil	Furnaces and boilers	21	.104	.018	.26	.44	.14	.0004
Gas	Furnaces and boilers	21	.618	.031	.16	Negl.	.02	.0007
Oil	Stationary combustion sources	22	--	.038	.54	.65	.08	.015
Gas	Stationary combustion sources	22	--	.075	.19	.001	.01	.011
Oil	Residential oil burners	23	.12	.008	.09	.29	.05	--
Oil	Industrial oil burners	23	--	.01	.12	2.15	.10	--

(a) Data given as percent CO.

TABLE 10. EMISSION DATA FOR EXTERNAL COMBUSTORS ASSOCIATED WITH STIRLING ENGINES AND STEAM ENGINES[(a)]

Fuel	Type of Equipment	Reference	CO, %	HC, ppm	NO_x, ppm	SO_x ppm
No. 2 diesel	GM Stirling engine -10 hp[(b)] with combustion air preheater	24	0.008	2	550	--
Kerosine	Williams steam engine	--[(d)]	0.05	20	70	--
JP-4	Thermo-Electron steam engine	--[(d)]	0.001	--	110	4
	Steam engine; developers name withheld by request	--[(d)]	0.3	30-40	25-35	--
Diesel fuel	Philips Stirling engine -80 hp[(c)] with exhaust-gas recirculation	--[(d)]	0.017	--	38	--

(a) When comparing the data in this table with those for gasoline engines on a relative mass-emission-rate basis, the data in this table should be roughly doubled to account for the higher air/fuel ratio generally used with external combustors.

(b) Operated at 25:1 air/fuel ratio.

(c) Operated with 50 percent of combustion air from recirculated exhaust gases.

(d) Data supplied by organization developing engine.

complete combustion of the fuel. Continued increase in the amount of excess air, however, can result in lower peak combustion temperatures which lead to less NO_x. Higher air/fuel ratios can also result in less complete combustion and consequent higher emission of HC and CO owing to reduced peak flame temperature. However, combustors specifically designed for fuel-lean mixtures need not suffer this disadvantage, since a primary combustion zone operating near stoichiometric can be used.

Figure 3 shows a typical variation of NO_x, HC, CO, and SO_x with air/fuel ratio. These data were obtained on an experimental test furnace designed to be typical of domestic oil-burning equipment.(16) An air/fuel ratio about 2.3 times the stoichiometric ratio yields minimum HC and CO emissions for this unit. Higher air/fuel ratios result in an increase in HC and CO emissions. The NO_x emissions increase almost directly with increasing air/fuel ratio to a maximum at 3.6 times the stoichiometric ratio. Beyond that point, the NO_x emissions begin to decrease. SO_x emissions increase with increasing air/fuel ratio until nearly all the sulfur in the fuel has reacted. For most external combustors, the amount of SO_x in the exhaust gases can be fairly accurately predicted from the sulfur content of the fuel.

Figure 4 presents some additional evidence of the relationship between air/fuel ratio and oxides of nitrogen. These data represent industrial and commercial burners ranging in heat-release rate from 100,000 to 4,000,000 Btu/hr, using natural gas and heating oil for fuel.(15) Although there is considerable scatter in the data points, the curves follow the same pattern as the NO_x curve on Figure 3.

Figures 5 and 6 show the variation of NO_x, HC, and CO with inlet air temperature and air/fuel ratio for a 10-hp(s) Stirling engine with a combustion-air preheater running with No. 2 diesel fuel.(24) Figure 5 shows that increasing the inlet air temperature has about the same effect on emissions as increasing the air/fuel ratio does. The higher inlet air temperature results in higher peak combustion temperatures which promote more complete combustion as well as higher concentrations of NO_x. Higher air/fuel ratios resulted in lower NO_x, HC, and CO emissions as shown in Figure 6. It is seen that data for this combustor differed from those given for other combustors in regard to the effect of air/fuel ration on NO_x formation.

No emission data were found for external combustors operated on gasoline, the alcohol fuels, LP gases, hydrogen, and ammonia. The emissions from gasoline and the alcohol fuels will probably be similar to the emissions from kerosine and JP-4. The LP gases would probably produce emissions similar to natural gas. The emissions from hydrogen and ammonia would consist primarily of oxides of nitrogen. Hydrogen burns with a relatively high flame temperature and therefore might be expected to produce higher NO_x emissions than would other fuels at the same operating conditions. Ammonia burns with a lower flame temperature and thus would be expected to produce lower NO_x emissions than would the other fuels. Of course, CO and HC emissions would not exist with these latter two fuels.

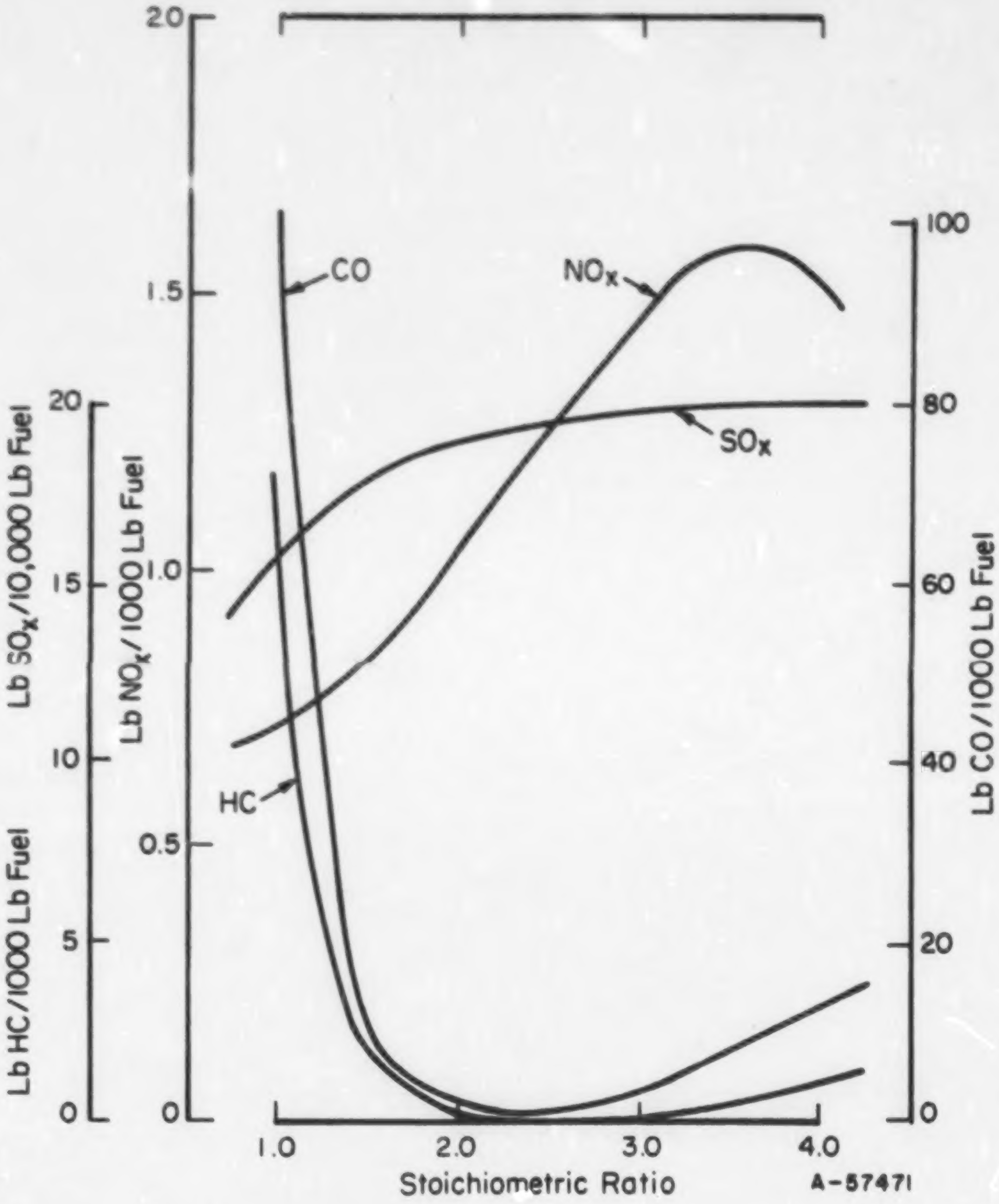

FIGURE 3. VARIATION OF EMISSIONS WITH AIR/FUEL RATIO FOR EXPERIMENTAL DOMESTIC OIL-BURNING FURNACE

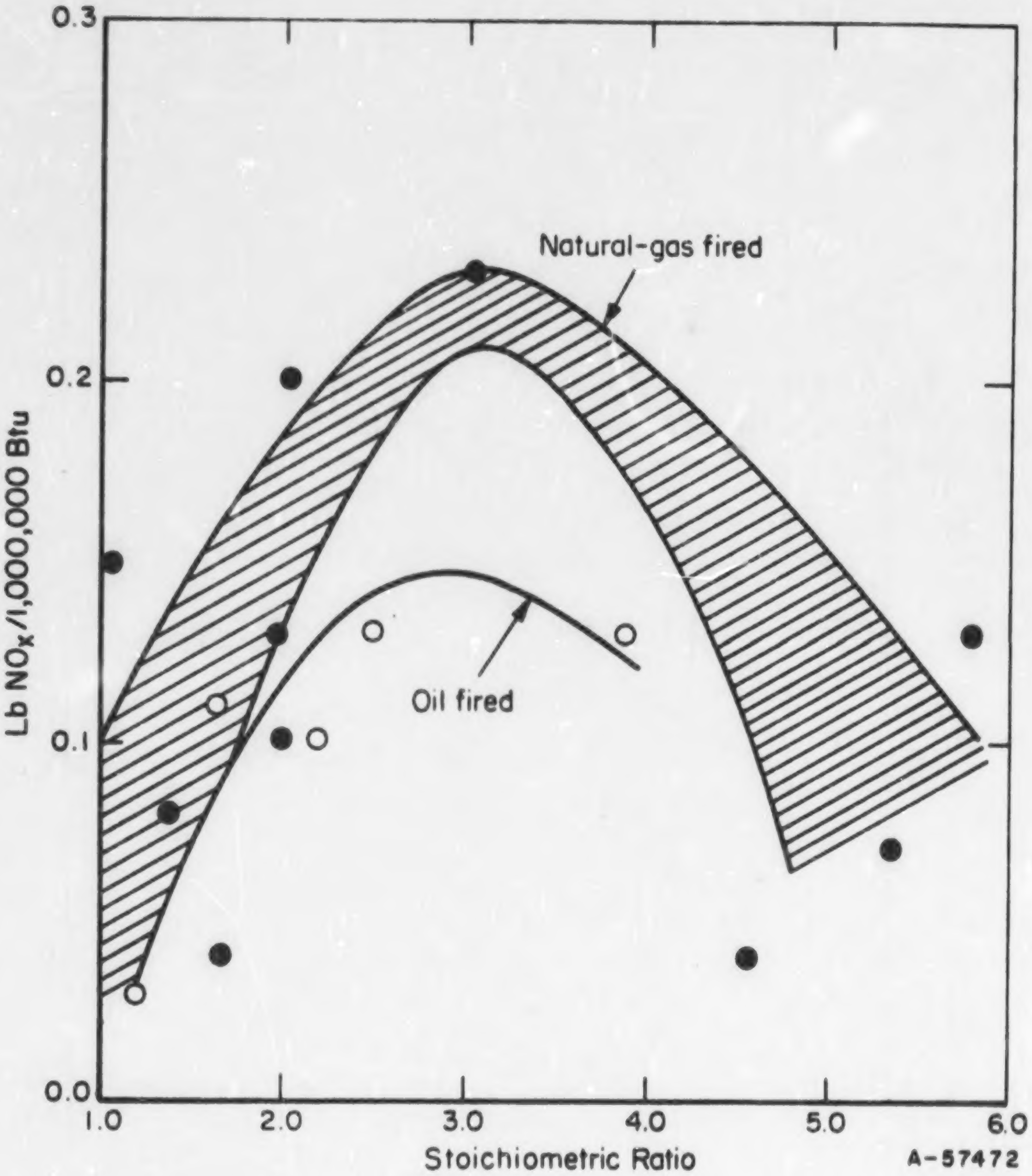

FIGURE 4. NITROGEN OXIDES EMISSIONS RELATED TO AIR/FUEL RATIO FOR NATURAL GAS AND OIL-FIRED IDUSTRIAL AND COMMERCIAL BURNERS

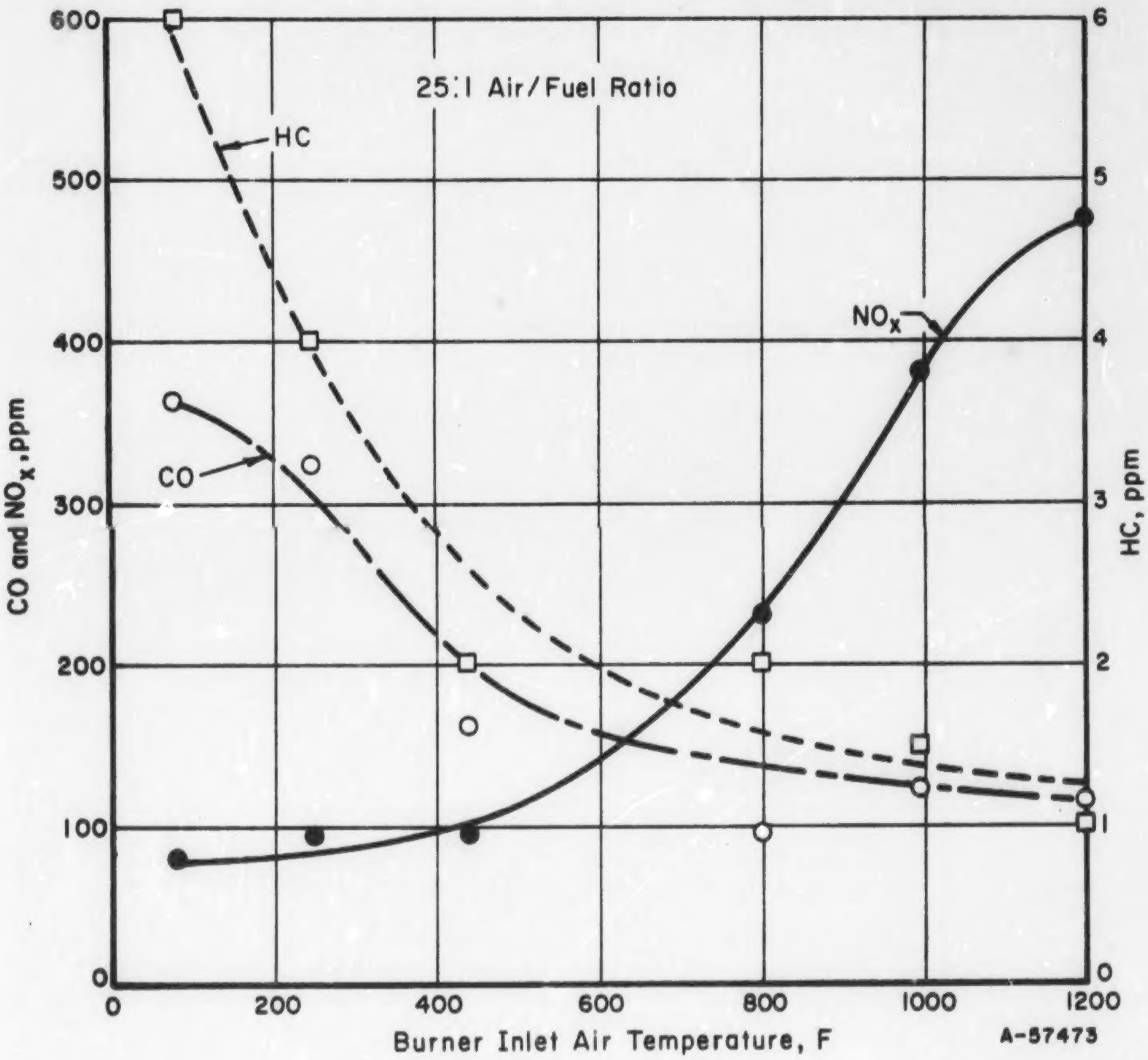

FIGURE 5. VARIATION OF EMISSIONS WITH INLET AIR TEMPERATURE FOR GM STIRLING ENGINE

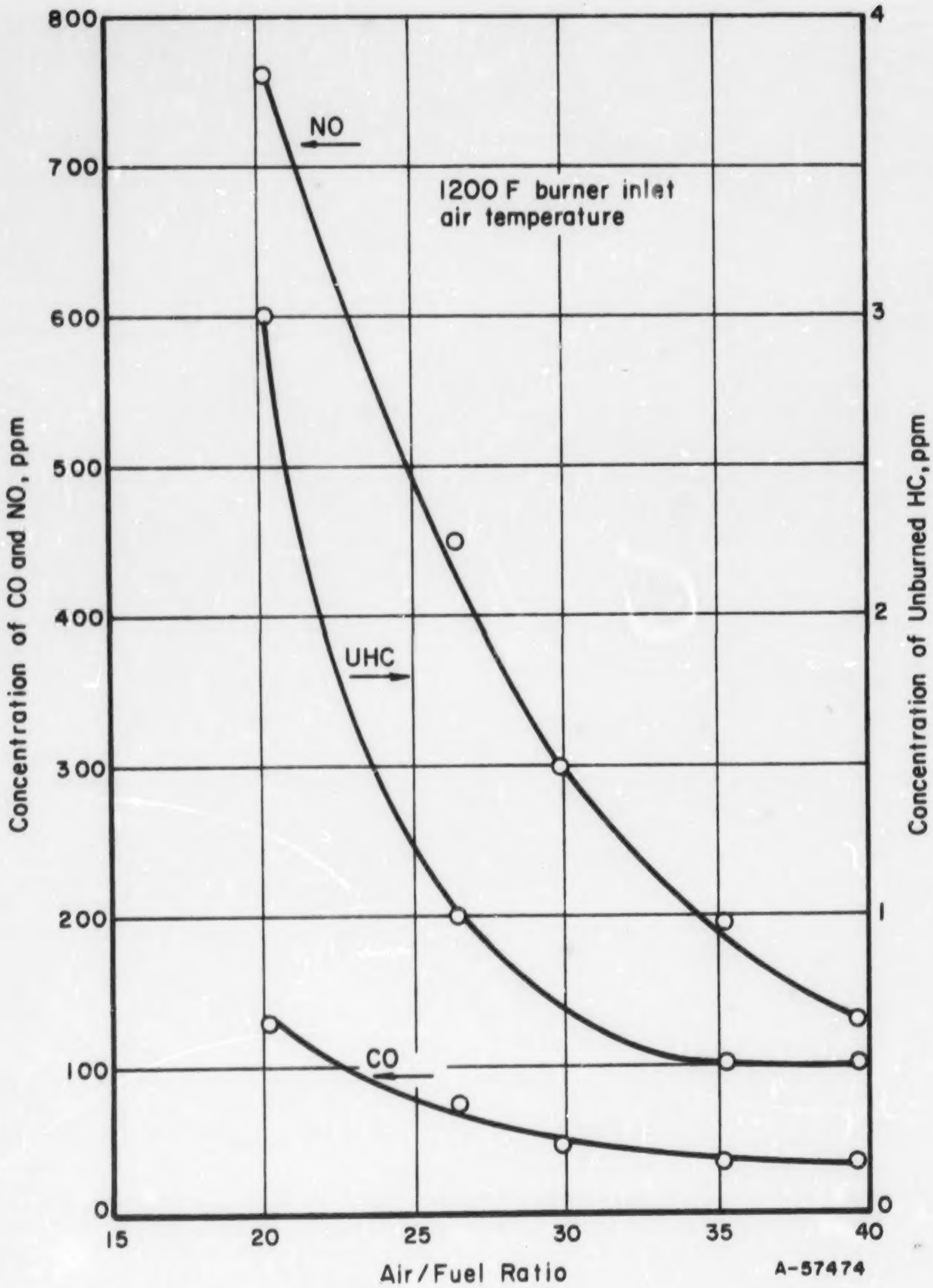

FIGURE 6. VARIATION OF STIRLING-ENGINE COMBUSTOR EMISSIONS WITH AIR/FUEL RATIO

HEAT ENGINES

BRAYTON-CYCLE (GAS-TURBINE) ENGINES

General Description

In its simplest form, the gas-turbine engine consists of a compressor, a combustion chamber, and a turbine. Air is taken in by the compressor at atmospheric pressure, compressed to a higher pressure, and then delivered to the combustion chamber where fuel is sprayed in and burned. The combustion process takes place at essentially constant pressure. The resulting high-temperature, high-pressure gas is expanded in the turbine and then exhausted to the atmosphere. Part of the shaft work developed by the turbine is used to drive the compressor. The remainder is the output work. A schematic diagram of the simple turbine engine is shown in Figure 7. The basic thermodynamic cycle on which this engine operates is called the Brayton cycle.

Several modifications of the simple cycle have been employed in the development of gas-turbine engines for vehicular use. In particular, all current engines for on-highway vehicles have a free power turbine. This is to say that two mechanically independent turbine stages are used: one to drive the compressor and the other to drive the output shaft. This allows a wide range in output-shaft speed which is not possible with a single-shaft turbine engine. A schematic diagram of a free-turbine engine is shown in Figure 8.

To improve efficiency, most automotive turbine engines use a regenerative cycle which incorporates a heat exchanger that delivers heat from the exhaust gas to the compressed air upstream of the combustor. This cycle is shown schematically in Figure 9. The heat exchanger is called a recuperator if it has a fixed surface, or a regenerator if it is a rotary type. Both types are used. The regenerator or recuperator also decreases the exhaust temperature, but results in a substantial increase in engine weight and volume.

One recently developed automotive-type gas-turbine engine uses a variable-slip coupling between the gas-generator section and the power turbine, as shown in Figure 10. This device is used to produce more optimum part-load cycle conditions. In addition, it provides excellent engine braking.

All of the gas-turbine engine types described above are open-cycle engines; that is, the working fluid is atmospheric air taken in by the compressor, and the turbine exhaust is discharged back into the atmosphere. Closed-cycle engines are another class of gas turbines. In a closed-cycle system, the working fluid is a high-pressure gas which is continually recycled, as shown in Figure 11. Heat is added to the working fluid through an additional heat exchanger instead of directly by a combustor. In most current closed-cycle developments, the intended source of heat energy is a nuclear reactor, although several fossil-fuel closed-cycle turbines have been built. A precooler is also required, to dissipate the heat that would ordinarily be rejected in the exhaust if the engine were open-cycle.

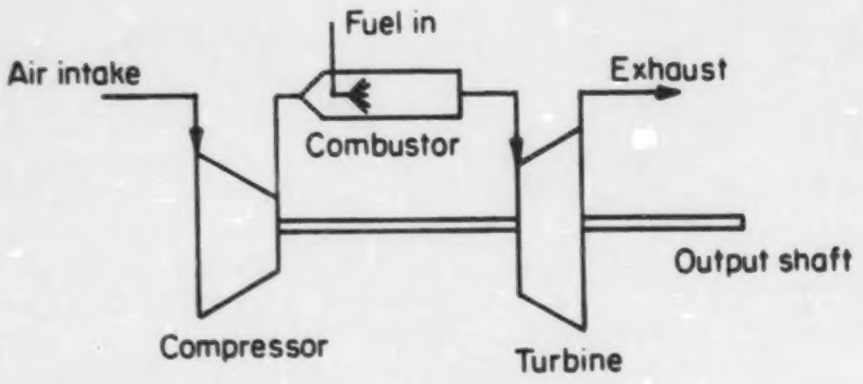
Fuel in
Air intake
Exhaust
Combustor
Output shaft
Compressor
Turbine

FIGURE 7. SCHEMATIC DIAGRAM OF SIMPLE GAS-TURBINE ENGINE

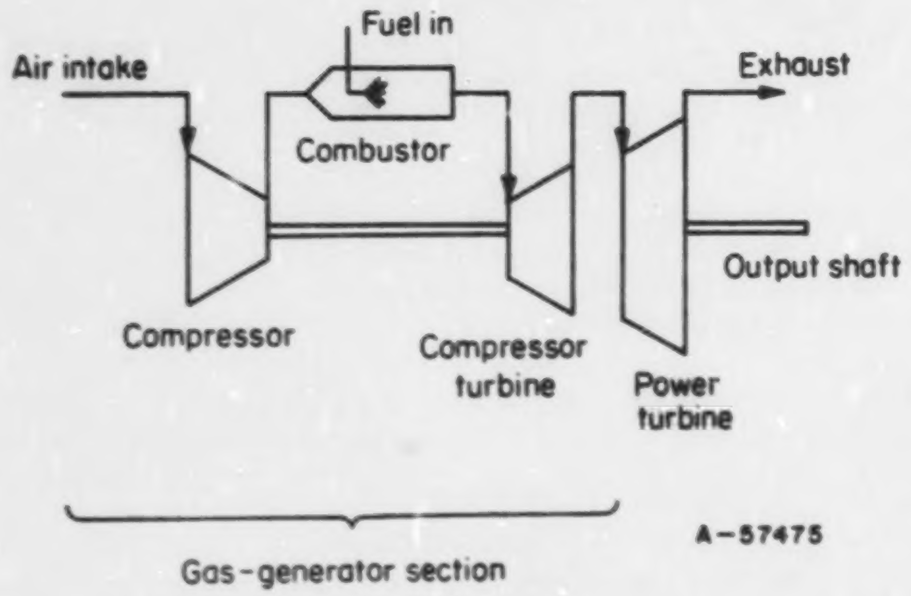
Fuel in
Air intake
Exhaust
Combustor
Output shaft
Compressor
Compressor turbine
Power turbine
Gas-generator section
A-57475

FIGURE 8. SCHEMATIC DIAGRAM OF FREE-TURBINE ENGINE

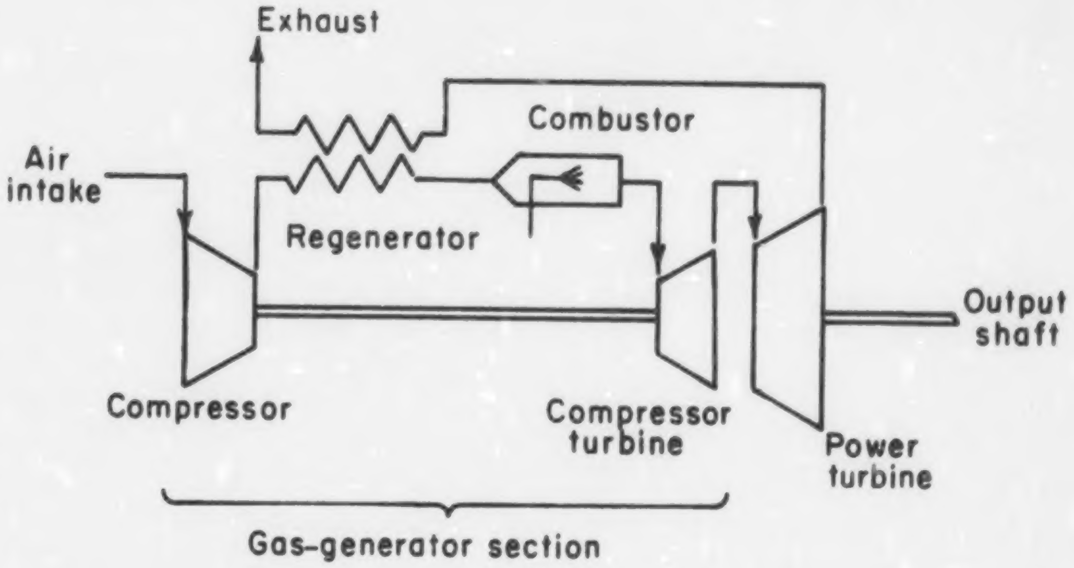
Exhaust
Combustor
Air intake
Regenerator
Output shaft
Compressor
Compressor turbine
Power turbine
Gas-generator section

FIGURE 9. SCHEMATIC DIAGRAM OF REGENERATIVE FREE-TURBINE ENGINE

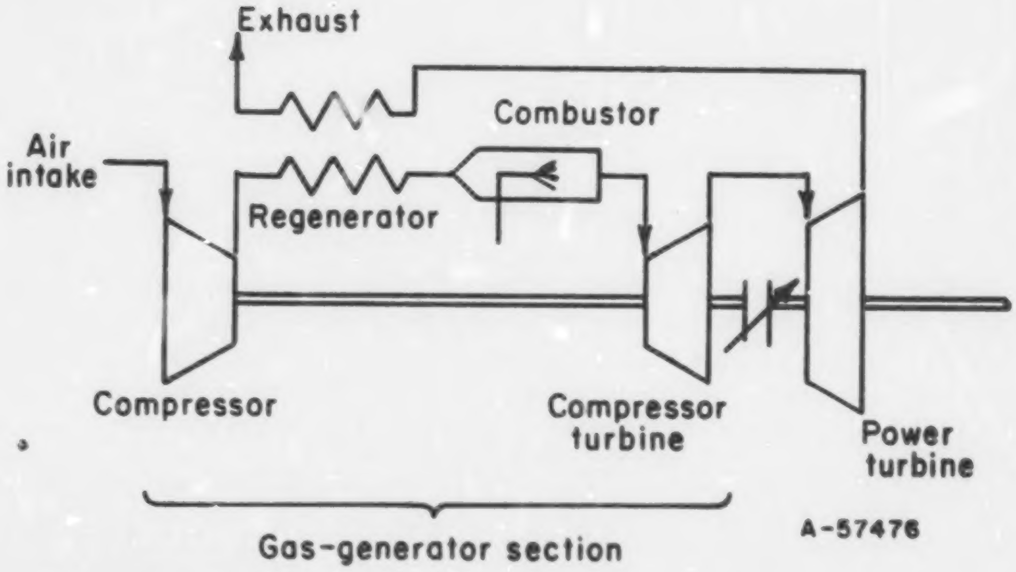
Exhaust
Combustor
Air intake
Regenerator
Compressor
Compressor turbine
Power turbine
Gas-generator section
A-57476

FIGURE 10. SCHEMATIC DIAGRAM OF REGENERATIVE FREE-TURBINE ENGINE WITH VARIABLE-SLIP COUPLING

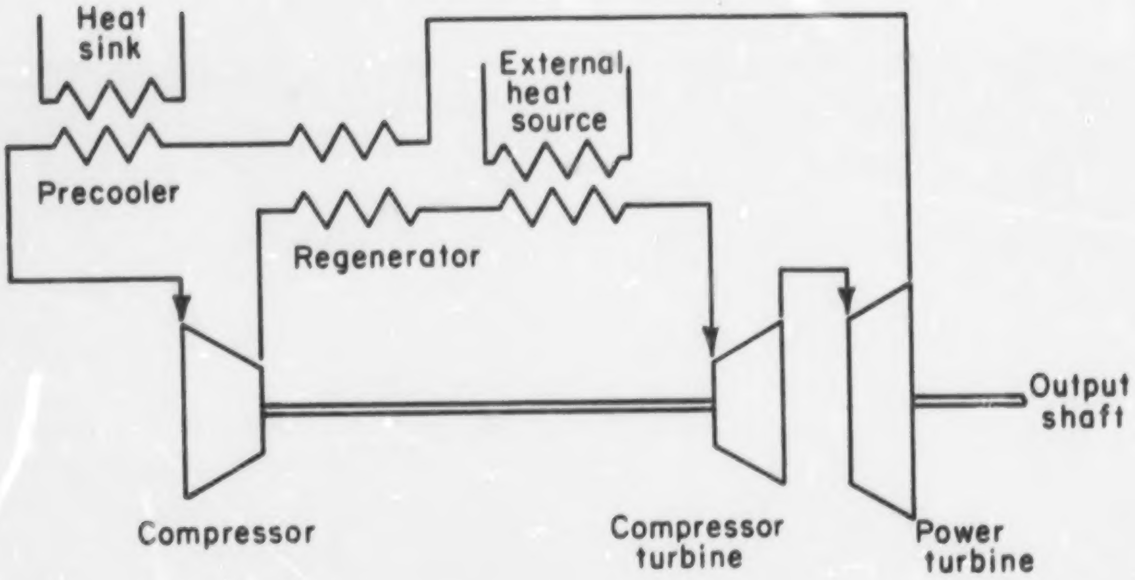

FIGURE 11. SCHEMATIC DIAGRAM OF CLOSED-CYCLE, REGENERATIVE, FREE-TURBINE ENGINE

Many other cycle concepts have been proposed and investigated. These include 3-shaft engines with intercooling between compressor stages and reheat between turbine stages, engines with differentially connected gas-generator, power-turbine, and output shafts, and engines having combinations of these features.

Historical Development

The fundamental principles of the gas-turbine engine were well known by the end of the eighteenth century; however, turbomachinery operating successfully on the Brayton cycle is a fairly recent development. The first really successful gas-turbine powerplant was the aircraft turbojet that developed from intensive work begun in the 1930s. These units first flew in the early 1940s and were fully operational in military aircraft by the late 1940s. The early developers had to learn how to design compressors and turbines of high efficiency and had to find suitable high-temperature materials. Rapid progress in the development of the aircraft turbine has come as the result of massive research efforts made in response to the obvious need for high-performance power plants for military aircraft.

The development of gas-turbine engines for automotive use was begun immediately after World War II, and has proceeded at a more modest rate of effort. Chrysler in this country and Rover in Great Britain were among the first to begin developing an automotive turbine. The first Rover turbine-powered automobile ran in 1949. General Motors began work on the automotive gas turbine around 1948, and Ford around 1950. All these developers selected the free-turbine configuration early and moved quickly to the regenerative configuration.

Parallel development efforts on small turbines for nonautomotive use were launched by Boeing, Solar, AiResearch Division of the Garrett Corporation, Lycoming, Continental, and Caterpillar. Most of these engine developments were based on the single-shaft, nonregenerative turbine.

Principal early automotive-turbine development problems were noise, fuel economy, durability, and acceleration lag. Acceleration lag proved to be one of the more difficult problems. For a free-turbine engine to increase power output, it is necessary for the gas-generator section to speed up. This takes time; hence, there is an acceleration lag as compared with conventional piston engines. Early automotive turbine engines took about 5 seconds from idle to develop full power, which resulted in unacceptable throttle response for passenger-car use.

In 1961, the Department of Defense sponsored a competitive design and development program for a 600-hp vehicular gas-turbine engine. The fuel-economy target in this program was a minimum bsfc of 0.4 lb/hp-hr, which is equivalent to 34 percent maximum thermal efficiency. Solar, Ford, and Orenda were the principal participants in this program. All three received development contracts with the Orenda development largely financed by the Canadian Government. While 0.4 lb/hp-hr fuel consumption was not achieved, sufficiently low fuel-consumption characteristics were demonstrated to encourage continued efforts.

In 1963, the Chrysler Corporation undertook a 2-year, 50-car consumer-evaluation program in which 50 turbine-powered automobiles were loaned for 3-month periods each to randomly selected families throughout the United States. The fact that such a program could be conducted bore testimony to tremendous progress. Chrysler never published in detail on the results of this program, but in press conferences, it was brought out that the drivers considered the turbine car "completely useable". Acceleration lag was considered by some to be still a problem. In general, the turbine car's fuel economy was poorer in the city, better on the open highway, and on the average, equivalent to that of a modern V-8 gasoline engine.

Current and Projected State of the Art

From the standpoint of physical and performance characteristics, the automotive gas-turbine engine appears to be an acceptable power plant at present; that is, present experimental turbine engines could be installed and would perform in conventional passenger cars and trucks, in a manner that would be acceptable to the consumer.

In passenger-car applications, the consumer might find the turbine a little noisier and a little higher in fuel consumption, but not prohibitively so. Acceleration lag would probably be perceptible but not overly disturbing. Idle-to-max-power acceleration times under 1 second are now being obtained. Under some conditions, an exhaust odor might be noticed. On the other hand, the consumers would gain advantages such as smoother idling, quicker warmup, freedom from cooling-system performance problems, and potentially lower maintenance.

The 2-shaft, regenerative gas turbine would have no particular weight and size advantage over the gasoline engine and at the present would have an initial cost disadvantage for passenger-car application. However, the turbine has one advantage to the public: turbine exhaust is substantially lower in CO and HC than are current gasoline engines of equivalent power.

There is considerable interest in the turbine's lower weight in the high-horsepower (over 300 hp) heavy-duty engine category. One current experimental 375-hp gas-turbine engine weighs 1700 lb, compared to at least 2750 lb for a conventional diesel of the same horsepower. In addition, the turbine does not require a radiator. As power requirements of commercial vehicles continue to increase, the incentive to use gas-turbine engines will also increase.

Durability and maintenance requirements of the automotive turbine have yet to be evaluated fully; however, the turbine can be considered potentially acceptable and possibly superior to conventional piston engines in regard to these factors. In commercial aircraft service, maximum time between overhauls for turbojet engines is now as high as 12,000 hours, as opposed to 3000 hours for aircraft reciprocating engines. In at least one instance, no time between overhauls is even specified with measurements of certain engine performance parameters being relied on to indicate if and when maintenance is required. The maintenance labor required for the turbojet is about 1/2 man-hour per hour of flight time, while that for the reciprocating engine is about 1-1/2 man-hours per flight hour. There is some reason to doubt that the regenerative automotive turbine can enjoy this same degree of superiority over the automotive piston engine; nevertheless, the aircraft experience is an attractive precedent. Chrysler has stated that no major durability problems resulted during the 50-car program.(26)

Cost is the principal factor that has yet to be evaluated. Reduction of manufacturing cost is probably the area of work receiving the greatest effort at present. There are two separate camps in regard to cost objectives. In the United States, Chrysler Corporation has the only development effort aimed at passenger-car applications. Chrysler is optimistic about eventually being able to produce a turbine engine for this application for the same cost as and having similar efficiency to a conventional gasoline engine giving the same performance. On the other hand, General Motors, Ford, and others are directing their developments toward large commercial-vehicle applications. In this effort, their avowed target is to meet the manufacturing cost and efficiency of the diesel engine of equivalent power.

These objectives are not inconsistent. Since the passenger-car power plant seldom is required to develop full power and has a short life requirement by commercial-vehicle standards, considerable economies can be made in the design and materials of the passenger-car engine.

Even though the automotive gas-turbine engine has been under development now for 20 years, progress in improving the performance of experimental engines is still advancing rapidly. Many improvements of present engine concepts can be foreseen but have yet to be made, and many advanced engine concepts have been identified which have only begun to be evaluated. For example, advances in cooled turbines for aircraft use, which permit the use of higher turbine inlet temperatures, have not yet been applied to automotive turbines.

Current Experimental Automotive Units

Table 11 lists some of the principal features of current experimental automotive turbines. Also listed for comparison are two experimental turbines for off-highway and nonautomotive applications. All but the last two are regenerative engines. It is probable that the performance figures in Table 11 are not all given for the same atmospheric conditions; therefore, the table is not intended to produce a precise comparison, but rather is intended to give an indication of the general levels of performance being obtained.

Chrysler Corporation. Chrysler released a detailed description of their fourth-generation gas-turbine engine in 1964.(27-30) This is the engine that was installed in the 50-car-program vehicles. The manufacturers claim this 130-hp(s) unit will provide vehicle activity equivalent to that of a large V-8 gasoline engine. The fact that automotive gasoline engines are rated without accessory loads or inlet and exhaust restrictions, while the turbine rating is presumably "as installed", helps to make this claim credible. Also, the torque of the turbine engine increases as output speed is decreased; therefore, the turbine can deliver near-peak power over a broader speed range than the gasoline engine can.

The Chrysler engine is a free-turbine regenerative unit using twin rotary disk regenerators, one on each side of the engine, rotating in the vertical plane. This engine also has variable-angle power-turbine nozzle vanes. At part load, these vanes rotate so as to reduce the flow area of the nozzle. This has the effect of increasing the compressor-turbine inlet temperature, and part-load thermal efficiency is thereby increased. The variable vanes can also be reversed to provide engine braking during vehicle deceleration.

The gas-generator acceleration time between idle and 97-1/2 percent full speed was initially 1.5 seconds; this has been reduced subsequently to less than 1 second.

Chrysler is now in the process of developing a fifth-generation engine. No plans to place the turbine engine in quantity production have been released by Chrysler.

Rover 2S/140. The Rover and Chrysler engines are the two principal examples of current engines intended for passenger-car use. Details of the 2S/140 were released in 1963.(31) The two engines are similar in size and basic arrangement, yet there are significant differences. Both use a centrifugal compressor, but the Rover unit incorporates a variable-inlet-guide-vane feature. The Chrysler engine has a fixed-geometry axial-flow compressor turbine, while the Rover uses a variable-nozzle-vane radial-inflow compressor-turbine configuration. The combination of the variable-geometry compressor and compressor turbine allows the Rover unit to idle at a higher speed, thereby reducing time required for gas-generator acceleration.

The Rover engine has an axial-flow power turbine, as does the Chrysler; as of 1963, the variable power-turbine-nozzle-vane feature was still under development for Rover.

TABLE 11. CURRENT GAS-TURBINE ENGINES

Developer	Model No.	Rating, hp(s) at rpm	Minimum bsfc, lb/hp(s)-hr	Maximum Thermal Efficiency	Weight, lb	Length, in.	Width, in.	Height, in.	Specific Weight[(a)], lb/hp(s)	Specific Volume[(a)], ft^3/hp(s)
				Automotive Turbines						
Chrysler	Fourth generation	130 at 3600	0.54	0.25	410	35	26	28	3.2	0.110
Rover	2S/140	150 at 5300	0.55	0.25	470	38	27	30	3.2	0.120
General Motors	GT-309	280 at 3600	0.45	0.31	950	36	30	36	3.4	0.081
Ford	707	375	--	--	1700	40	32	39	4.5	0.077
			Turbines for Off-Highway Vehicles and Nonautomotive Applications							
Caterpillar/ Boeing	553-2	400	0.75	0.18	385	39	27	24	1.0	0.037
USAMERDC/ AiResearch	331-30	500 at 6000	0.60	0.23	275	34	33	24	0.6	0.031

(a) Includes reduction gearing and integral accessories; excludes transmission and air cleaner.

The Rover 2S/140 engine employs twin stationary recuperators to provide regeneration. However, a subsequent unit, the 2S/150/R, uses glass-ceramic twin rotary-disk regenerators, very similar to Chrysler's in shape and location.(32) The 2S/150/R and the 2S/140 presumably use the same basic turbomachinery components.

This basic engine has been used several times in the LeMans 24-hour race, with good durability records.

General Motors GT-309. The GT-309 engine uses a centrifugal compressor and axial-flow compressor and power turbines, all fixed geometry.(33) This is the only engine to date using the variable-slip coupling between the gas-generator and power-turbine sections, which General Motors calls "Power Transfer". Part-load fuel economy is enhanced by this feature; in addition, it provides excellent engine-braking capacity.

The GT-309 regenerator is a drum type, presumably ceramic, rotating in the horizontal plane and located on the top of the engine.

The General Motors engine is intended for use in commercial vehicles and, at 280 hp(s), is in a power range where it is in direct competition with diesels. The unit has been installed in a GMC tractor and tested with gross combined weight up to 78,000 lb. At 65 mph, the GT-309 delivers the same miles per gallon as a diesel of equivalent power, but at lower speeds the diesel is more efficient.

The GT-309 is the fifth gas-turbine engine developed by General Motors Research Laboratories. GM's turbine work has recently branched out to the GM Detroit Diesel Division, where production versions of the engine are reportedly being designed.(34) Target dates for production have not been announced. Initial units will reportedly be intended for industrial rather than vehicular applications.

Ford 707. No performance figures have been released on Ford's most recent turbine development, except that it is designed to produce 375 hp(s).(35) This places the 707 near the top end of the power range now covered by truck diesels. The weight saving is considerable over a diesel of comparable power.

The engine appears very similar in configuration to the Chrysler engine, with a centrifugal compressor, axial-flow turbines, variable power-turbine nozzle vanes, and twin rotary-disk regenerators. Two previous Ford turbine engines used a complicated 3-shaft arrangement that had excellent potential for good part-load economy, but which has apparently been dropped.(36)

Ford officials announced last year that they intend to produce gas-turbine engines for trucks by the "early 1970s". This might be a little optimistic; nevertheless, it appears that Ford has a substantial turbine program under way.

Caterpillar/Boeing. The Caterpillar Tractor Company recently purchased rights to the gas-turbine designs and technology that were developed by Boeing. The Boeing unit is included in Table 11 as an example of the level of efficiency and specific size and weight that can be obtained with a free-power-turbine nonregenerative unit. It is evident that the regenerative feature adds considerably to bulk and weight; however, the efficiency of nonregenerative units is generally unacceptable for commercial use. Nevertheless, the Boeing unit is a vehicular engine and has been used in military vehicles and for marine propulsion. The most recent Boeing experimental engines are reported to be regenerative.

Caterpillar has had a turbine-development program of its own for a number of years before the Boeing purchase.(38) Caterpillar is reportedly interested in producing turbine power plants for over-the-highway commercial vehicles.

USAMERDC/AiResearch 331-30. This engine is being developed for the United States Army Mobility Equipment Research and Development Center (USAMERDC) by AiResearch and is currently in the final phases of development. It is a single-shaft, nonregenerative machine and is included in Table 11 as an example of the level of efficiency and excellent specific size and weight that are possible with this kind of machine.

Lycoming AGT-1500. The development of a 1500-hp(s) automotive-type regenerative gas-turbine engine was announced in 1967 by Avco Lycoming Division and the U. S. Army Tank-Automotive Command.(39) Design targets are 1600 lb weight and a fuel consumption of 0.38 bsfc. This is equivalent to a specific weight of about 1 lb/hp(s) and a thermal efficiency of 36 percent.

This engine is intended for use in heavy military vehicles, and is mentioned here as an example of future performance levels that can be expected from very large engines. Lycoming is a large manufacturer of turbines for helicopter use.

Other Developments. Other organizations are capable at present of producing automotive-type gas turbines. Both Solar and AiResearch have developed regenerative engines and both have an incentive to be interested in vehicular engines. Solar has been a Division of International Harvester for some time, and AiResearch and Mack Trucks are now both subsidiaries of the Signal Oil organization.

Continental has been both a developer of turbine engines and a producer of vehicular engines for some time, and should be regarded as a potential producer of automotive gas-turbine engines.

Closed-Cycle Gas-Turbine Engines

Closed-cycle gas turbines have not been seriously considered as an automotive power plant. The principal disadvantage of the closed-cycle system is the large additional heat exchangers that would be required to transmit the heat to the working fluid in place of the combustor, and to reject the heat that normally goes out the exhaust. The required precooler alone would probably be three times the size of a conventional automobile radiator. A second disadvantage is the internal lubricant sealing problem. Since conventional lubricants leaking into the working fluid would decompose thermally and foul the system, it has been necessary to use either gas-lubricated bearings or elaborate sealing systems in current experimental nuclear-powered units.

Closed-cycle Brayton engines are of interest principally for the conversion of nuclear power – both reactor and radioisotope – and solar energy. The Army has had a 400 hp(e), portable, reactor-powered system under development for several years.(40) This system uses nitrogen gas as the working fluid.

Smaller Brayton-cycle units ranging from 4 hp(e) to 27 hp(e) have been investigated by NASA and USAF for aerospace applications.(41, 42) These units use either argon or helium-xenon mixtures as the working fluid. Gas-lubricated bearings are used in the turboalternator units.

At present, closed-cycle gas-turbine engines do not appear to have any potential for automotive use.

Practical Size Range

From available data on current experimental automotive units, it appears that gas-turbine engines are certainly practical in the range of 100 hp(s) and above. However, turbines do not miniaturize very well. As design horsepower is reduced, the design speed of turbine engines increases and the diameter of the rotating components gets smaller. Very small rotating components usually suffer from high ratios of tip clearance to blade height, which tend to degrade component efficiency. In addition, the cost competitiveness of the turbine suffers as design horsepower is reduced.

The Williams Research Corporation has built a 70-hp(s) regenerative turbine engine which delivered good efficiency installed in a pleasure boat. This organization has also built a 30-hp(s) engine that ran at 85,000 rpm with only slight penalties in component efficiency. It appears, then, that a 30 hp(s) vehicular gas turbine may be practical from a performance standpoint, but would probably suffer a cost disadvantage. It is not believed, however, that a competitive vehicular gas turbine of the low capacity, 16 hp(s), required for the utility-car application could be developed.

Fuel Requirements

Gas-turbine engines will tolerate a broad range of fuels, and operate best on fuels such as kerosene, JP-4, No. 1 diesel, and No. 2 diesel. Unleaded gasoline can be used, but it is not as desirable as a turbine fuel. Chrysler Corporation, in the 50-car program, recommended that unleaded gasoline be mixed with diesel fuel in warm weather, or used by itself in cold weather.(43) Turbine engines should not be run for extended periods on leaded fuels because the combustion products cause fouling and corrosion.

Air Filtration

Entrained dust particles can cause erosion and rapid deterioration of the rotating components of unprotected turbine engines; therefore, efficient air filtration must be provided. Because turbine engines consume large quantities of air and are sensitive to inlet pressure losses, the required filter units are fairly large – comparable in size to the radiator of a conventional automobile. The air-filtration problem has been investigated in some detail by the U. S. Army Tank-Automotive Command and by USAMERDC and has been found to be manageable in terms of military operational requirements.(44) Passenger-car requirements are probably less severe.

Emission Characteristics

To date, the data on gas-turbine-engine exhaust emissions are somewhat limited; however, it is apparent that the gas-turbine exhaust can be substantially cleaner than exhaust from present controlled-emission gasoline engines in regard to unburned hydrocarbons and carbon monoxide. Also, no lead compounds are present in the turbine exhaust, as unleaded fuels are used.

General Motors has recently released data comparing emissions from their GT-309 gas-turbine engine and an equivalent, controlled-emission gasoline engine on the California Cycle.(45) These data indicate that less than 1 ppm unburned hydrocarbons and less than 50 ppm (0.005 percent) carbon monoxide were present in the turbine's exhaust over a broad range of operating conditions. These figures cannot be compared directly with existing gasoline-engine standards, however, because the total mass of gas-turbine exhaust is 10 to 100 times higher than that of a gasoline engine of equivalent power, depending upon the mode of operation.

On a total-mass-flow basis, the GT-309 emitted only about 16 percent of the hydrocarbons and 12 percent of the CO that were emitted by the controlled-emission gasoline engine with which it was being compared. Since the California test emission rating for that gasoline engine was 222 ppm hydrocarbons and 1.28 percent CO, the emissions from the turbine would be equivalent to those of a comparable gasoline engine having an emission rating of approximately 36 ppm hydrocarbons and 0.15 percent CO. This level of emission is only 4 to 5 percent of that which is typical for noncontrolled-emission gasoline engines.

Also, a comparison of total mass of emissions is not necessarily valid either. It can be conjectured that when the GT-309 engine is operating in an environment in which the ambient concentration of hydrocarbons is 1 ppm or above, as is sometimes the case, and if the turbine engine emits less than 1 ppm in the exhaust, then the turbine is actually cleaning up the environment. Therefore, it appears that the net contribution to atmospheric pollution is the meaningful parameter. Further, it is apparent that net contribution is meaningful only when compared with an acceptable standard ambient pollution level.

In regard to oxides of nitrogen, the GM data show that the GT-309 emits a greater total mass of NO_x than the comparable piston engine – in fact, 1.75 times as much. The computed composite emission value for the particular gasoline engine in question is about 1620 ppm. Thus the equivalent value for the GT-309 is about 2800 ppm NO_x. Ford and Chrysler, on the other hand, indicate that emission of NO_x from present engines can be below projected standards of 350 ppm, on a corrected basis. Apparently, the turbine does not enjoy a clear advantage with respect to NO_x, although emission levels can be at least potentially acceptable for the near future.

In regard to smoke and odor, two manufacturers claim no problem with either, and a third admits to having problems with both under some operating conditions.

Anyone who has stood close to a taxiing jet aircraft and watched a jet take off knows that gas-turbine exhaust can exhibit an objectionable odor and can contain smoke. Also, it is known that a poorly designed or defective burner of any type can deliver poor combustion. It is doubtful that gas-turbine researchers have a complete understanding of how burner-design variables relate to exhaust-emission levels. Therefore, it seems reasonable to expect some significant differences in exhaust-emission levels between different engines, and also to expect improvements to be made.

One final consideration with respect to emission characteristics is the fact that gas turbines have very low oil consumption and the level of oil consumption does not tend to drastically increase with age as with gasoline engines. This will result in less deterioration in emission characteristics with age.

"1980" Physical, Performance, and Cost Characteristics

Specific Weight. It is doubtful that the weight characteristics of gas-turbine engines for automobiles will be improved substantially in the next few years. While technological advances and design improvements may make weight reductions possible, cost-reduction considerations will probably require weight-increasing compromises such as the use of lower strength or higher weight, but lower cost, materials. A specific weight of 3.0 lb/hp(s) including air cleaner is projected for a "1980", 180 hp(s) engine. While there is no representative 30 hp(s) engine upon which to base a projection of specific weight, it is believed that the specific weight of a "1980", 30 hp(s) engine would be approximately 4.0 lb/hp(s).

Specific Volume. The specific volume would be on the order of 0.09 ft^3/hp including air cleaner for a 180-hp(s) gas-turbine engine. Again, while there is no representative 30 hp(s) engine upon which to base a projection, it is believed that the specific volume of a "1980", 30 hp(s) engine would be approximately 0.12 ft^3/hp(s). Of the above volumes, required air-cleaner volume would be on the order of 0.01 ft^3/hp(s), including ducting.

Efficiency. A maximum brake thermal efficiency of 35 percent should be attainable with a 180 hp(s) engine, equivalent to a bsfc just under 0.4 lb/hp(s)-hr. At 30 hp(s), some penalty can be expected – perhaps as maximum of 30 percent efficiency could be obtained. Part-load efficiency characteristics are shown in Figure 12. The curves shown are representative of three types of engines, one with variable-angle power-turbine nozzle vanes, another with a variable-slip coupling between the gas generator and power turbines, and the third with variable-geometry compressors and turbines. These curves are drawn with the assumption that the power turbine is running at peak-efficiency speed for the given power level. The Rover engine characteristic is considered typical of what could be achieved in "1980".

Startup Characteristics. The start-up time of a gas-turbine engine is dependent upon the capacity of the starting system and will probably be less than 15 seconds to power output under normal ambient conditions. Warm-up time for full power is more rapid than for a conventional piston engine. The starting sequence is as follows: the cranking motor turns the gas-generator section up to ignition speed at which the fuel is turned on and lit off. The cranking motor continues to assist up to some speed below idle at which the turbine is self-sustaining. In all probability, starting systems will be fully automatic, and the sequence will be initiated by the driver simply turning the "ignition" key.

Power-Output Characteristics. Although peak power-turbine speeds will be in the range of 20,000 to 50,000 rpm, reduction gearing is used to reduce the output-shaft speed to conventional IC-engine speeds.

Typical speed-torque characteristics are shown in Figure 13. As is evident from this figure, the torque of the turbine engine rises as the output-shaft, or power turbine, speed is reduced. Since there is no fixed coupling between the free power turbine and the gas-generator section, the engine will operate with the output shaft stalled. In the example shown in Figure 13, the stall torque is over three times the maximum-speed torque, although the ratio of stall torque to maximum-power torque is about 2.2. Since the power-turbine and output shaft can be stalled while the gas-generator section continues to run, and since the engine has a large torque increase at stall, the use of a fluid coupling, or torque converter is not required in an automatic transmission designed for use with the gas turbine.

Corresponding typical speed-power characteristics are shown in Figure 14. At a given gas-generator speed, the power output is dependent upon the output-shaft speed. The dashed line is drawn through the peaks of the power curves and represents an optimum-load characteristic for free-turbine engines. Fortuitously, vehicle road-load speed-power curves are similar in shape, and, therefore, near-peak economy during cruising can be obtained with a single high-gear ratio.

As mentioned previously, the gas-turbine engine cannot deliver a sizable instantaneous step increase in power, since the gas generator must change speed to significantly change power in present turbines. However, it has been demonstrated that full-acceleration response time can be brought below 1 second. Since gasoline engines have carburetion and induction lags on the order of a few tenths of a second, the turbine acceleration lag is probably not distressing, if even perceptible. Engines using a variable-geometry compressor and compressor turbine could be controlled so as to change air flow and hence, power, without change in gas-generator speed, thereby further improving the response characteristics of the turbine engine.

Power-Surge Capability. The gas turbine does not have a power-surge capability in excess of maximum power.

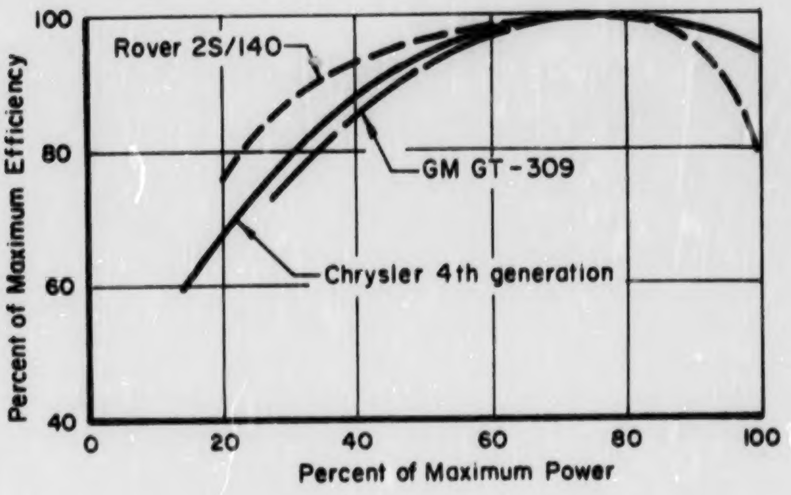

FIGURE 12. PART-LOAD EFFICIENCY CHARACTERISTICS OF REGENERATIVE GAS-TURBINE ENGINES

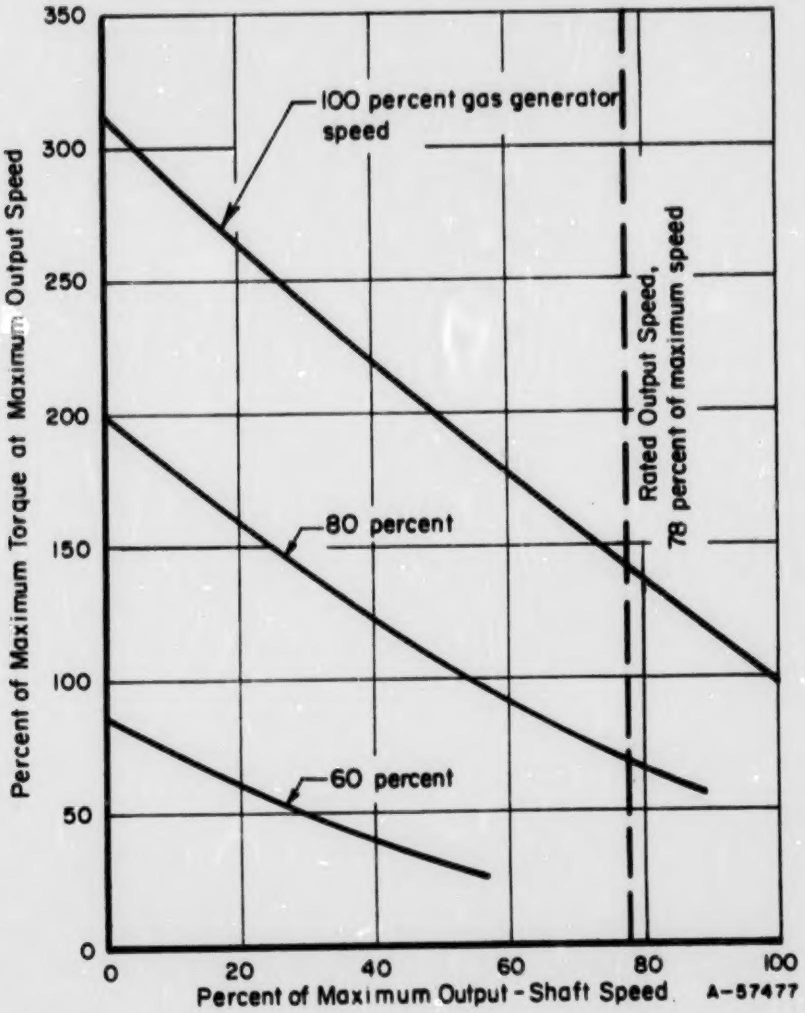

FIGURE 13. SPEED-TORQUE CHARACTERISTICS OF A FREE-TURBINE AUTOMOTIVE GAS-TURBINE ENGINE

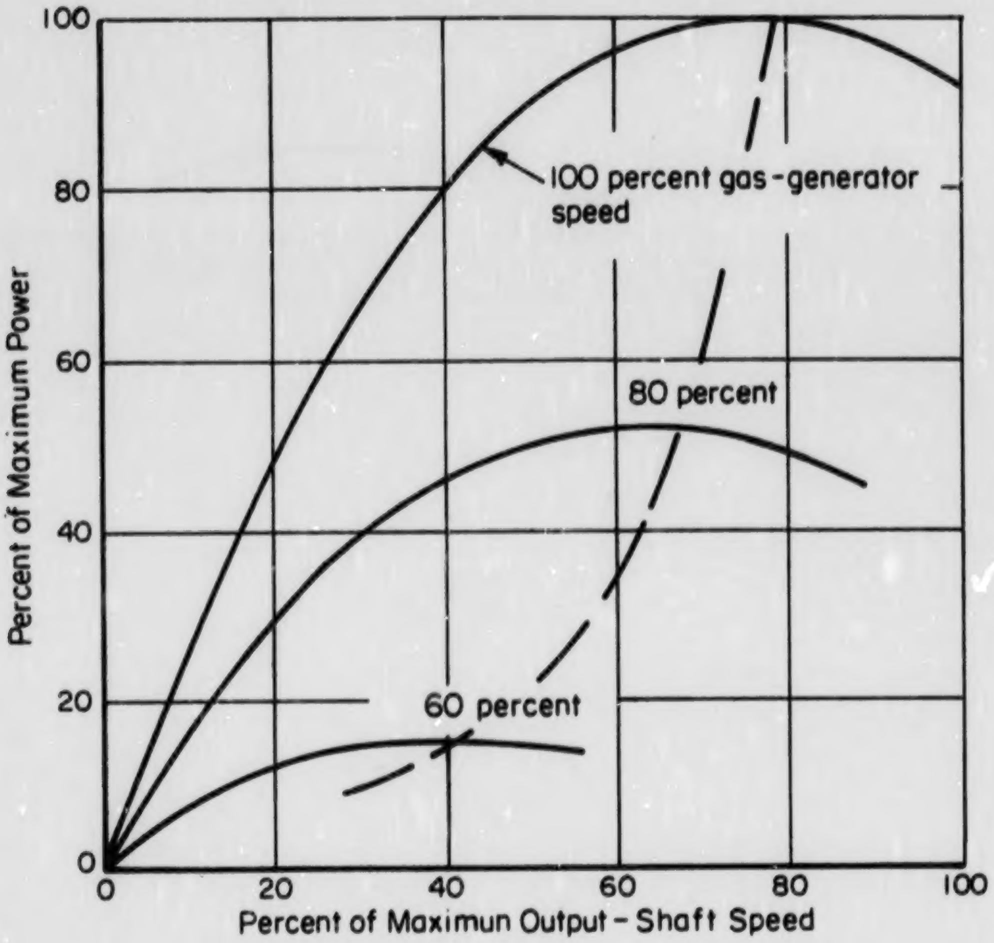

FIGURE 14. SPEED-POWER CHARACTERISTICS OF A FREE-TURBINE AUTOMOTIVE GAS-TURBINE ENGINE

Specific Cost. If produced in quantities similar to those of passenger-car gasoline engines, and if designed for similar life and operating conditions, at least one major developer believes that a passenger-car turbine can potentially cost about the same as a present-day gasoline power plant in the 180 hp(s) size, which may be interpreted as approximately 2 to 3 \$/hp(s). Assuming that cost per pound increases slightly as engine parts become smaller, the turbine might cost 3 to 4.5 \$/hp(s) at 30 hp(s). On the other hand, a high-efficiency, long-lived, lower-production-quantity, truck or bus turbine would probably cost about the same as an equivalent diesel truck engine, or 15 to 20 \$/hp(s).

RANKINE-CYCLE (STEAM) ENGINES

General Description

The Rankine-cycle engine is an external combustion engine in which high-pressure steam or some alternative working-fluid vapor is expanded in either a turbine or a positive displacement (i.e., piston-type) expander to produce work. Figure 15 shows the components of a typical Rankine-cycle engine in schematic form. The high-pressure liquid pump draws liquid at a low pressure from the reservoir and forces it under high pressure through the heater where the liquid picks up heat from the expander exhaust vapor. For some working fluids and/or engine applications, this liquid heater is not used or required. The high-pressure liquid then enters the vapor generator where it is converted to superheated vapor by further heating. The hot, high-pressure vapor is then metered into an expander through a flow-control valve. The vapor expands to a low pressure and temperature and gives up energy to the expander. The vapor leaves the expander and enters the liquid heater where it gives up more heat energy. The vapor is then converted back to a liquid in a condenser, which is typically air cooled as shown. The resulting low-pressure liquid is then returned to the reservoir by a pump to complete the fluid cycle.

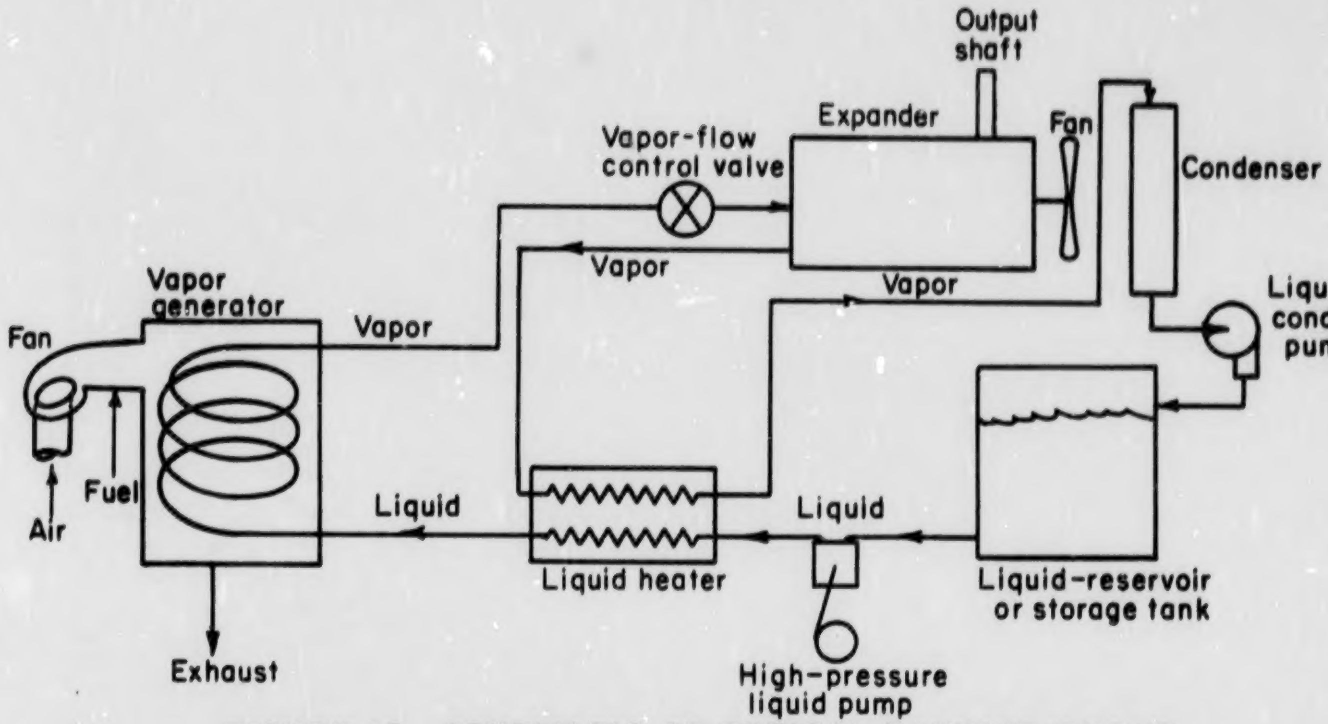
Output shaft
Expander
Fan
Vapor-flow control valve
Condenser
Vapor
Vapor
Liquid condensate pump
Vapor generator
Fan
Vapor
Fuel
Air
Liquid
Liquid
Liquid heater
Liquid-reservoir or storage tank
Exhaust
High-pressure liquid pump

FIGURE 15. SCHEMATIC OF TYPICAL RANKINE-CYCLE ENGINE COMPONENTS

Historical Development

The use of Rankine-cycle engines in vehicles dates back to 1827, or before, when primitive steam engines were used to power coaches. Steam cars were sold commercially in the late 1800s and early 1900s with familiar names such as Stanley Steamer, Locomobile, White, and Doble. Of the 126 different makes of steam cars produced, the

Stanley Steamer was the most popular. Most of the early steam automobiles, like the Stanley, had a boiler containing a large quantity of water, and used a noncondensing, single-expansion engine. With the large thermal-storage capacity of the boiler, up to 1/2 hour was required to get going from a cold start. Since they were noncondensing, these engines also used large quantities of water. The Stanley Steamer was produced from 1899 through 1925, reaching a peak production of 2,500 vehicles in 1910. A condenser in the system was introduced in 1915.

The Stanley Steamers were quiet-running, lightweight cars of surprisingly good performance, even by present standards. Although maximum continuous speeds were limited to 45 to 60 miles per hour by the low heat-release rate of the boiler, tremendous bursts of acceleration and high speed were possible because of the large energy-storage capacity of the boiler.

Perhaps the most advanced vehicle built during the steam era was the Doble Steam Car, produced by Abner Doble during the twenties. The latest Doble was built in 1930. The vehicle had a 142-inch wheelbase and weighed 5,500 pounds. The 4-cylinder steam engine ran condensing and was rated at 150 hp(s), continuous. The engine had a low-inertia monotube boiler that was electrically controlled and could get up to pressure within 30 seconds.

Initially, the gasoline auto gained popularity because it was free from the frequent water stops and difficult boiler-control problems of the steam car. However, the ultimate demise of the steam car was apparently not due to lack of continuing engineering development but to lack of adequate service and repair facilities, and to the high costs resulting from low-volume production, as compared with the more popular gasoline car.

In 1951, the McCulloch Corporation undertook the development of a modern automotive steam power plant for installation in a luxury sports-type car. Although many advantages were claimed for the engine, no units were ever sold, and the program was dropped in 1954.

In 1964, the Convair Division of the General Dynamics Corporation published a design study of a steam-turbine power plant for a military battle tank. From this study, it was concluded that the steam power plant was well suited to the tank application, as well as to other off-road vehicular applications. Design-optimization and cost studies remained to be done at that time, however.

Apparently, the only automotive steam powerplant that has been offered for general sale in recent years is made by the Williams Engine Company of Ambler, Pennsylvania. In 1967, the Williams brothers were taking orders for their complete power plants for $6,450, or a Chevelle automobile with their engine installed for $10,250. These engines were to be made in lots of ten after accumulation of sufficient orders.

Actual development and/or design studies of automotive steam powerplants are being conducted by other organizations at present. Thermo Electron Engineering Corporation (TEECO) of Waltham, Massachusetts, has developed a sealed, 0.13 hp(e) portable steam-powered generator unit. More recently, TEECO has prepared preliminary paper studies of automotive units. Controlled Steam Dynamics of Mesa, Arizona, has reportedly been engaged in the development of both a small unit of a few horsepower output and an automotive-size steam power plant. Details of this development are reported to be forthcoming in 1968. Other developments receiving publicity recently include: Gibbs & Hosick of Winston-Salem, N. C., with several swash-plate piston expanders ranging up to 200 cu in. displacement; Smith and Petersen of Midland City, California, with a steam engine for installation in a Volkswagon and a 42 cu in. outboard engine converted to steam; and Pritchard Steam Power, Ltd., of Victoria, Australia, with a 100-hp, V-2 steam engine in a 5-ton truck.

The development of steam power plants for other applications, such as large electric utility generation plants, marine propulsion units, and railroad locomotives are well known. One perhaps lesser-known development was a unit built by the Besler Corporation for aircraft propulsion that flew for short periods in 1933.

Recently, Rankine-cycle engines have received some consideration for use in small, silent, military ground power units and for aerospace energy-conversion applications. The work on the small ground power units has been sponsored principally by the United States Army Mobility Experimental Research and Development Center (USAMERDC) at Fort Belvoir, Virginia. Various experimental units have been built using steam, mercury, and organic fluids(46) as the working fluid. One such unit is the 4-hp(e) SCAP system, which is a turbine-drive, mercury, Rankine-cycle generator set developed for USAMERDC by TRW(47).

Units for aerospace application are being developed principally for the conversion of solar energy, radioisotope power, or reactor power to electrical energy. Developers include TRW, Philco-Ford, Aerojet-General, and AiResearch. Working fluids for the systems are primarily liquid metals or organic fluids.(48)

Current and Projected State of the Art

Automotive experience with Rankine-cycle (steam) engines of recent design is very limited. However, presuming that the application of modern technology to the design of Rankine-cycle engines can produce durable, trouble-free systems, the Rankine-cycle engine must be regarded as a potentially quite acceptable passenger-car power plant.

The operating characteristics of positive-displacement Rankine-cycle (steam) engines are likely to produce strikingly different sensations to the driver or passenger accustomed to gasoline-engine-driven cars. One of the principal differences is that the steam engine stops whenever the vehicle stops; there is no need to unclutch and "idle" the engine. During acceleration, manual changes in the cutoff position will produce about the same sensation as gear changes in a conventional automatic transmission. If a continuously variable automatic cutoff control is used, there will be no gear-change sensations whatever.

The principal noise from the steam engine is that emitted by the combustion system. Since the expander is relatively quiet, there will probably be little impression that the power plant is laboring under high-speed or rapid-acceleration conditions.

Current Experimental Automotive Units

General. All recent Rankine-cycle engines built for automotive use have steam for the working fluids and have piston-type expanders. The engines are semiclosed-cycle systems that require some make-up water. Boilers are low-thermal-inertia or "once-through" types, which give good response and which permit rapid warmup from a cold start. The pistons are lubricated by conventional hydrocarbon oils, which need to be separated from the feedwater to avoid fouling problems in the boiler. Freezing is still a problem in cold weather. It can be assumed that an objective in advanced developments will be to achieve completely sealed, self-lubricating systems.

Data on several automotive steam engines, either built or studied since 1950, are given in Table 12.

The Williams Engine. As previously stated, the Williams engine is apparently the only automotive steam power plant that has been offered for general sale in recent years. The expander is a 4-cylinder, single-acting, single-expansion crosshead machine of 105-cu in. displacement with a 3.50-in. bore and a 2.75-in. stroke. A number of standard gasoline-engine parts are used in construction of this engine. Each cylinder has an intake valve, an exhaust valve, and an exhaust port in the cylinder which is uncovered near the bottom of the piston stroke. The intake-valve motion is controlled by a camshaft having compound lobes which change the valve timing as the camshaft is moved axially into one of four positions.

With steam at 1000 psi and 1000 F, a power output of 150 hp(s) or 250 hp(s) is claimed for continuous or intermittent operation, respectively. Stall torque is 1105 lb-ft, and good fuel mileage is claimed – 20 to 30 mpg.

Steam is exhausted at 3 to 9 psig and 230 to 300 F. The steam passes through a shell-and-tube feedwater heater and then through a condenser, which is a heavy-duty conventional radiator. The condensate and any uncondensed steam is pumped by a gear pump to a water reservoir which is vented to the atmosphere. Water loss through this vent can amount to 12 gal in 500 miles. Water is pumped into the boiler through the feedwater heater by a 3-cylinder piston pump. Water flow is controlled by a boiler pressure control.

The steam generator is a stainless-steel monotube flash boiler heated by an on-off pressure-atomizing fuel burner which is controlled by a boiler temperature and pressure control. Normal operation from a cold start can be resumed in 30 seconds.

The pistons are lubricated by oil injected through the cylinder wall.

Turbine Versus Positive-Displacement Expanders

In the past, turbine expanders for automotive steam engines have not been attractive. There are two principal reasons for this. One is the large number of stages required to obtain efficient operation with steam turbines. The second is that in automotive applications, turbine expanders require a transmission, while piston-type

TABLE 12. DATA ON SEVERAL AUTOMOTIVE STEAM ENGINES EITHER BUILT OR STUDIED SINCE 1950

Developer or Researcher	Engine Reference	Rating, hp at rpm	Rated Pressure, psia Temperature, F	Specific Weight, lb/hp(s)	Specific Volume, ft^3/hp(s)	Maximum Efficiency, percent	Start-Up Time[(a)], sec	Torque Ratio[(c)]	Power-Surge Ratio[(c)]	Specific Cost, $/hp(s)
Williams Engine Co.	49	150 at 2400	1000 1000	5.4	0.14	--	PO<30	3.4	1.67	44
McCulloch Corp.	50	120 at 1200	2000 900	8.0	--	23	FPO<30	--	1.25	--
Gibbs Hosick Trust	51	60 at 2500	2000 850	--	--	--	--	2.3	--	--
Richard J. Smith	52	250 at 6000	1000 700	--	--	28	FPO-14	1.8	--	--
USAMERDC	--(d)	3 at 3600	700 850	20.0	--	19	PO-120	--	--	--
Thermo Electron Engineering Corporation	53(e)	100 at 1680	1200 1250	5.0	--	28	--	1.1	--	--
Microtech Research Co.	54(e)	175	1500 1100	17.2	0.67	16	FPO-500	--	--	--
General Dynamics/Convair	55(e)	500	1200 1000	--	0.16	22	--	--	1.5-2.0	--
S. W. Gouse, Jr.	25, 56(e)	--	--	5-10	--	25-30	--	--	--	3 (?)
Battelle/Northwest	57(e)	50	2500 670	--	--	24	FPO<10	--	--	--

(a) PO = time to power output; FPO = time to full power output.

(b) Torque ratio = the ratio of stall torque to rated torque.

(c) Power-surge ratio = the ratio of short-term, "burst", power to continuous rated power.

(d) Estimated parameters for steam engine currently under test by United States Army Mobility Equipment Research and Development Center. This engine is not for vehicular application, but is included to illustrate state of the art for small steam engines.

(e) Paper studies only.

expanders, in general, do not. The variable cutoff capability of the piston expander enables high stall torque to be generated and eliminates the need for gear changes. In addition, the rotation of piston expanders can be reversed without transmission gearing.

Nevertheless, a turbine expander would have a favorable torque characteristic, similar to that of a two-shaft gas turbine, with a stall torque about twice the maximum-power torque. Therefore, the steam-turbine transmission would be simpler than that required for a gasoline or diesel engine. Because of the inherent durability advantages of the turbine, as compared to piston expanders, Rankine-cycle turbine engines are potentially attractive for use in commercial vehicles such as trucks and buses, where higher power engines are required and where higher first cost can be justified to obtain longer life and lower maintenance costs. In addition, the use of higher-molecular-weight fluids, such as organic fluids, enables the use of fewer stages in the turbine expander. Should any of these fluids prove acceptable for use in automotive Rankine-cycle engines, the turbine expander would look more attractive.

Alternative Working Fluids

As discussed earlier, for some nonautomotive applications, liquid metals and certain organic fluids as well as water are being used as the working fluid in Rankine-cycle engines. These alternative fluids generally are being used in systems with turbine expanders.

The attractiveness of candidate fluids depends upon their chemical, mechanical, and thermodynamic properties, including long-term chemical stability at elevated temperatures and lubricating qualities. One problem with organic-fluid systems is that they are currently limited to relatively low vapor generator temperatures. This is because current candidate fluids generally are not chemically stable at temperatures above 600 to 700 F, although data on the long-term thermal stability of these fluids are far from complete. Some organic fluids, particularly those containing only carbon and fluorine, are reported to be stable at temperatures higher than these, but their overall desirability for use in Rankine engines has not yet been determined.

The potential for designing systems in which the working fluid can also act as a lubricant enhances the desirability of using the organic fluids in positive-displacement as well as in turbine systems. One additional attractive feature of the organic-fluid system is its freedom from the freezing problem.

A major problem with liquid-metal systems for automotive application would be their toxicity hazard.

In conclusion, because of water's good chemical, mechanical, thermodynamic, and thermal stability properties, particularly with respect to use in lower power, positive-displacement expander systems, it is currently the most likely candidate for passenger-car Rankine-cycle engines. If the lubrication and freezing problems encountered with water are not satisfactorily resolved, however, one of the organic fluids may prove desirable for this application. For vehicle applications where a turbine expander is desirable, such as where long life and maintenance-free operation are required, the use of fluids other than water appears to be desirable. In any event, much investigative work remains to be done with respect to alternative fluids before optimum fluids can be determined for various vehicle applications.

Emission Characteristics

The emission characteristics of Rankine-cycle engines would be consistent with those discussed earlier for the general class of external-combustion engines. Reported data for the Williams engine are: 20 ppm HC, 0.05 percent CO, and 70 ppm NO_x. The HC and CO emissions, while very low compared with those of gasoline engines, are not outstanding in comparison with those attained with other external combustors.

"1980" Physical, Performance, and Cost Characteristics

Specific Weight. The specific weight of the two existing automotive steam engines listed in Table 12 is in the range of 5 to 8 lb/hp(s). These specific weights include the weight of all components shown on the schematic in Figure 15. It is important to note that for automotive application, a transmission will probably not be required. Since many of the engines listed in Table 12 are design studies only, are in the earliest stages of development, or have some existing shortcomings that remain to be resolved, no reduction of this specific weight, 5 to 8 lb/hp(s), is estimated for fully developed, "1980" engines.

Specific Volume. The specific volume listed for the Williams steam engine is approximately 0.14 ft^3/hp(s). Assuming a higher condensing capacity and resolution of other existing shortcomings, an equivalent specific volume, 0.12 to 0.16 ft^3/hp(s), is estimated for fully developed "1980" engines.

Efficiency Characteristics. With accessory loads, heat losses, etc., the maximum reported brake thermal efficiency of current engines is around 28 percent. Future development work should increase this to about 30 percent.

Figure 16 shows the variation in brake thermal efficiency at typical road loading conditions claimed for two engines. These data should not be considered typical of steam engines in general but do show that efficiency holds up well down to low power levels. Engines should and will be tailored to give best economy for the applicable vehicular duty cycle.

Start-Up Characteristics. Current steam engines require under 30 seconds to develop power output from a cold start and somewhat over 30 seconds to full power. One design study on steam engines claims a start-up from a frozen condition will be possible in 10 seconds. The boiler will require more sophisticated controls as boiler size, thermal inertia, and start-up time are reduced. Such controls could be developed on the basis of current technology. The danger of damage from water in the cylinders during a cold start has been eliminated by suitable valving.

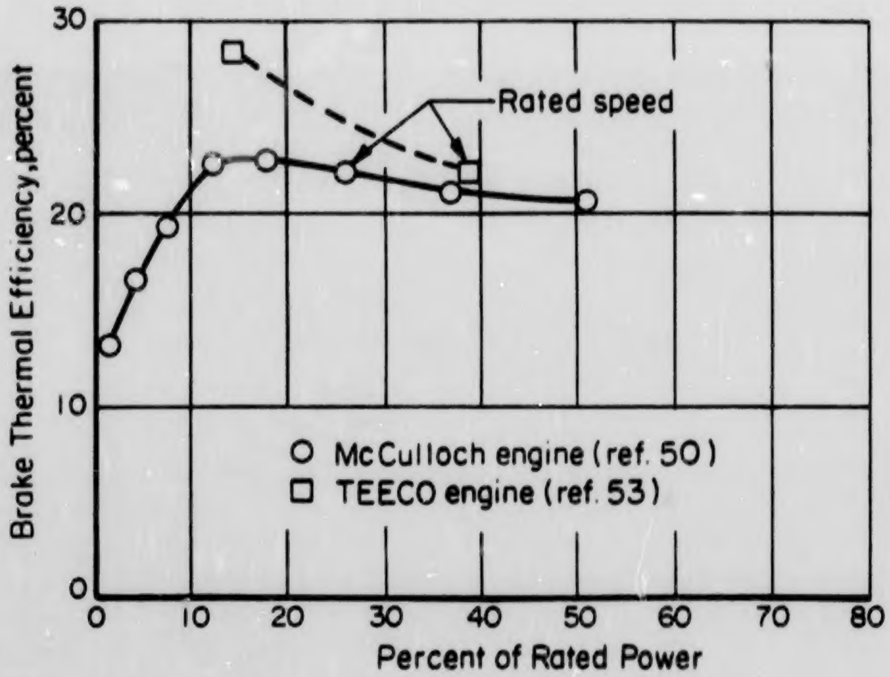

FIGURE 16. STEAM-ENGINE EFFICIENCY AT TYPICAL ROAD LOADING CONDITIONS

Power-Output Characteristics. Table 12 gives data on the rated power, speed, and torque ratio of pertinent engines. Figure 17 gives the torque and power as a function of speed in dimensionless terms for two engines. The difference in just these two designs reflects the great flexibility in output available with steam power. A typical reciprocating steam engine will have provisions for varying the cutoff point. The cutoff affects the amount of steam allowed to enter the cylinder and do work on the piston. Varying the cutoff effectively accomplishes the same thing as a gear shift in a transmission. Thus, no one dimensionless plot would be typical of steam engines in general.

Power-Surge Capability. Early steam cars had large power-surge capability because of thermal-energy storage in the boiler. However, as system weight, size, and start-up time are decreased, the thermal inertia of the boiler is reduced and the power-surge capability diminishes. One current engine which has about a 30 second start-up time has a short-duration power-surge capability of 167 percent of rated power. Another experimental vehicle engine had a power-surge capability of approximately 125 percent of rated power. Since future engines will have low-thermal-inertia vapor generators to minimize weight, size, and start-up time, an effective power-surge capability of 125 percent of rated power is assumed for the "1980" engine.

Specific Cost. It is impossible to fix a specific cost figure based on current data because no figures are available which reflect volume-production quantity. The best available evidence indicates that the specific cost will be less than that of a comparable diesel engine and about equal to that of an equivalent V-8 engine with automatic transmission. The additional cost of the transmission will be eliminated with steam power.

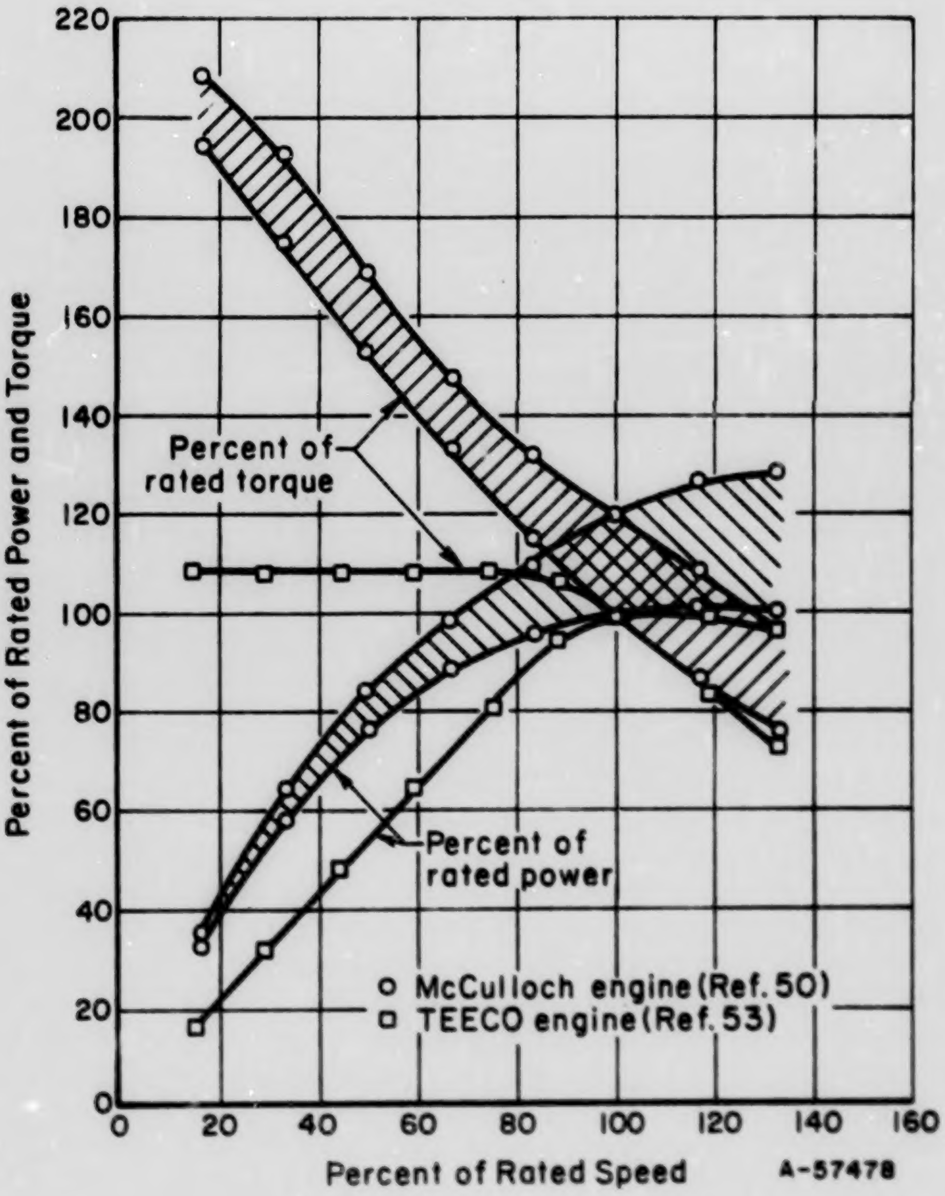

FIGURE 17. EXAMPLE STEAM-ENGINE TORQUE AND POWER CHARACTERISTICS

Assuming that: (1) a steam engine will cost about the same as an equivalent V-8 engine with automatic transmission; (2) current larger V-8 engines cost about $2 to 3/hp(s) and that smaller engines might cost $3 to 4/hp(s); (3) automatic transmissions cost about $1.0/hp(s); and (4) in equivalent installations the steam engine would be rated at about 20 percent less power than actual available V-8 engine power, an estimated cost of approximately $4 to 6/hp(s) is calculated for the steam engine.

STIRLING-CYCLE ENGINES

General Description

In its most common form, the Stirling-cycle engine, sometimes called a regenerative hot-air engine, is an external-combustion, closed-cycle, piston-type power plant that uses a gaseous internal working fluid, usually hydrogen or helium. In one currently preferred configuration, pistons are arranged in pairs and linked together through a common drive mechanism. One piston is called the displacer, and the other is called the power piston, as shown in Figure 18. Heat is added by an external combustor to the heater tubes connected to the cylinder on one side of the displacer piston. The water-cooled, cooler tubes are connected to the other side of the cylinder between the displacer piston and the power piston. The heater tubes and cooler tubes are connected to each other through the regenerator, which is usually composed of small cylinders filled with a fine wire matrix. At any instant, the internal working-fluid pressure is essentially uniform, since the working spaces formed by the two pistons are connected to each other through the cooler tubes, regenerator, and heater tubes.

In operation, the motion of the power piston lags the motion of the displacer piston by about one-quarter of a revolution. Referring to Figure 18, when the power piston is in the downward position, the displacer piston is moving upward. This forces the bulk of the working fluid around through the heater tubes, regenerator, and cooler tubes to the "cold" space below the displacer. In so doing, the bulk of the fluid is cooled and its pressure drops. The power piston then moves up, compressing the fluid and increasing its pressure. The displacer then moves downward, moving the bulk of the fluid back to the hot space and further increasing its pressure. The power piston then moves downward, expanding the fluid and completing the cycle. These motions do not occur separately, but are effected by the out-of-phase motion of the two pistons.

Because the internal fluid pressure is low when the power piston moves upward and high when the piston moves downward, a net work output is produced. The effect of the regenerator is to improve efficiency by storing heat from the hot fluid as it flows toward the cold space and releasing the heat when the flow reverses.

Fuel is burned in the combustion chamber shown at the top of the engine, and the combustion gases pass through the array of heater tubes surrounding the combustion chamber. The gases leave at a high temperature and, therefore, to conserve heat which would otherwise be wasted, exhaust gases pass through the preheater which cools the exhaust and warms the incoming combustion air.

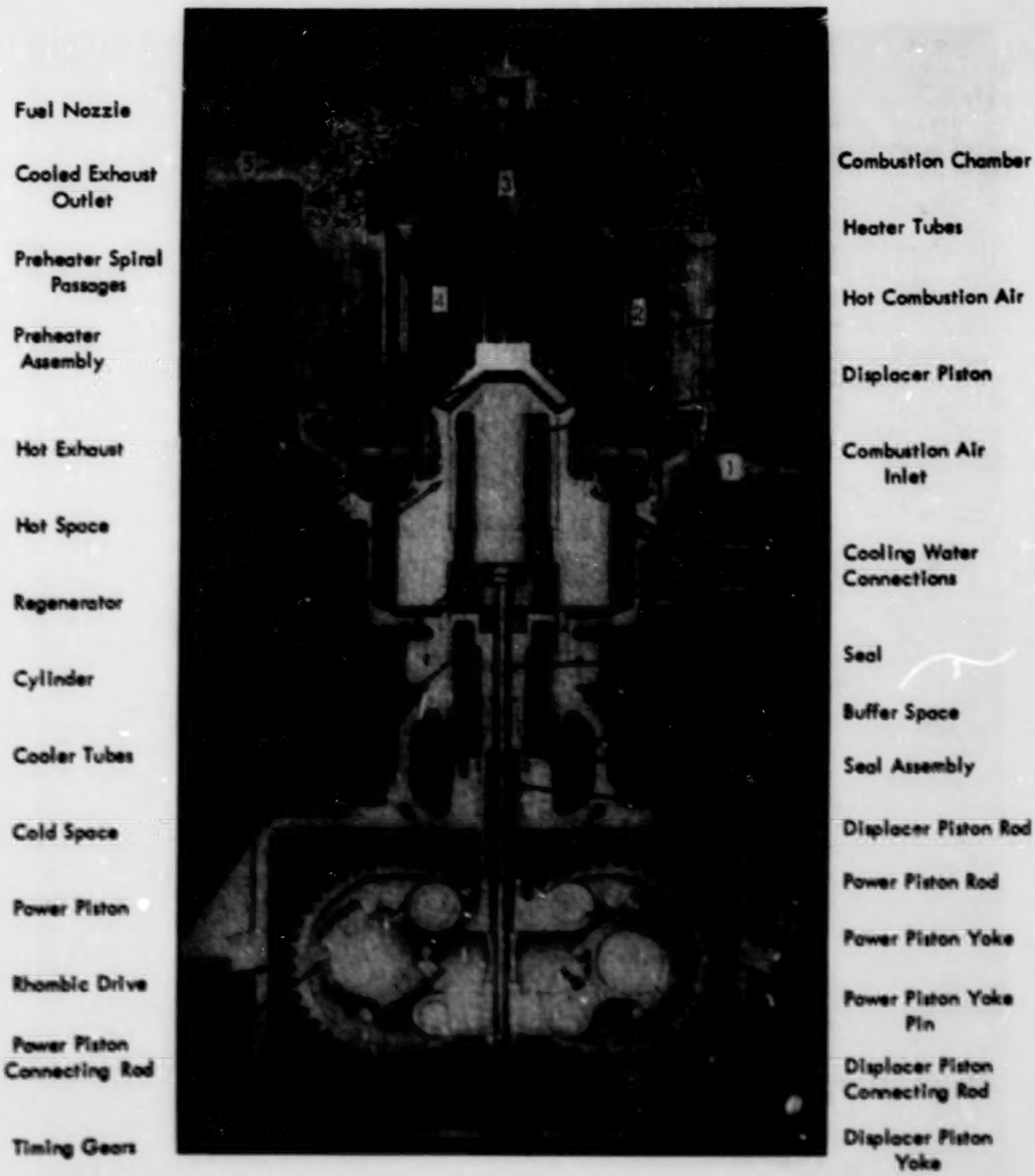
Fuel Nozzle
Cooled Exhaust Outlet
Preheater Spiral Passages
Preheater Assembly
Hot Exhaust
Hot Space
Regenerator
Cylinder
Cooler Tubes
Cold Space
Power Piston
Rhombic Drive
Power Piston Connecting Rod
Timing Gears
Combustion Chamber
Heater Tubes
Hot Combustion Air
Displacer Piston
Combustion Air Inlet
Cooling Water Connections
Seal
Buffer Space
Seal Assembly
Displacer Piston Rod
Power Piston Rod
Power Piston Yoke
Power Piston Yoke Pin
Displacer Piston Connecting Rod
Displacer Piston Yoke
5
3
4
2
1

FIGURE 18. CROSS-SECTION DRAWING OF DISPLACER-TYPE STIRLING ENGINE

Figure reprinted from Reference (58) by permission of General Motors Corp.

Historical Development

The Stirling-cycle engine is not a recent development. It was patented in 1816 by Robert Stirling, a Scottish clergyman. Thousands of engines operating on this principle were built and used in the nineteenth century, usually without the regenerative feature. These engines were slow and inefficient, but were desirable because they were substantially safer than the steam engines of that day. However, they were eventually made obsolete by the development of the more efficient and compact internal-combustion engines, and by the advances in the art of making explosion-proof steam-engine boilers.

Just prior to World War II, the N. V. Philips Laboratories at Eindhoven, Netherlands, undertook the development of a modern version of the Stirling engine as a means of producing electrical power in remote areas. This new breed of Stirling engines, characterized by high efficiency and quiet operation, has remained in the experimental stage through the present time. The principal developments in the U. S. have been carried out by General Motors Corporation under a cooperative licensing agreement with Philips. However, other organizations have shown interest and have built and tested experimental units. In addition, it is estimated that thousands of rudimentary hot-air engines have been built by hobbyists in basement workshops.

Current and Projected State of the Art

Until the present, applications of the Stirling-cycle engine have been limited because of some unsolved development problems that resulted in poor durability. However, as a result of continued development by Philips Laboratories and by various divisions of General Motors Corporation over the past 10 years, Stirling engines are now being built that will run for long periods of time. A few engines in the U. S. have been run for over 1000 hours. These engines are much quieter than IC engines, quite efficient, and are roughly equivalent to the present diesel engines in size and weight. Excellent throttle response is attainable, and exhaust emission from the external combustor is characteristically clean. Thus, the Stirling engine is considered to be a potentially acceptable vehicular power plant. In relation to present passenger-car engines, the primary factors limiting the use of Stirling engines are the relatively high first cost, weight, and size and the large radiator that is required. While the gasoline engine rejects most of its waste heat through the exhaust, the Stirling engine rejects most of its waste heat through the radiator.

Mechanical Configuration

To date, the combined work of Philips and General Motors has produced the only highly sophisticated Stirling-cycle power plants that have been publicized. The Stratos Division of the Fairchild-Hiller Corporation is known to have been conducting a Stirling-cycle development program; however, their developments have not been made public. The currently most popular mechanical configuration, shown in Figure 18, was developed by Philips. It is called a "displacer" type, as opposed to an alternative configuration called the Rider type, after a hot-air engine built in 1876. The Rider engine has two separate cylinders, but operates on essentially the same thermodynamic cycle as the displacer engine. Stirling's original engine was a displacer type. The unique feature of the Philips' configuration is the "rhombic" drive mechanism. Although somewhat complex compared with conventional crankshaft drives, the rhombic drive produces theoretically perfect balance of all inertial forces, and eliminates side-thrust on the pistons.

The power output of the Stirling engine can be controlled by raising or lowering the combustion temperature; however, this method is too slow for most applications because of thermal lag of the heater tubes. To achieve rapid response to power-demand changes, the output of the engine is modulated by changing the pressure level of the hydrogen or helium internal working fluid. To reduce power output, some of the gas is pumped out by a small diaphragm pump to a reservoir which is kept at elevated pressure. Conversely, to increase power, some gas is admitted back into the engine. Maximum mean pressure level inside the engine is usually about 1500 psia.

To achieve rapid power cutoff, a port between the "cold" space and the buffer space can be opened, which reduces the power-piston compression. Excellent throttle response is said to be obtained by this means.

Current Experimental Units and Applications

Experimental units have been produced by General Motors' Allison Division, Electromotive Division, and Research Laboratories, and by the N. V. Philips Laboratories of the Netherlands. Table 13 summarizes data on these engines. Perhaps the most widely publicized development has been the 10-hp(s) GPU-3 single-cylinder engine built by GM Research for evaluation by USAMERDC as a "silent" military ground power unit. This engine has passed 500-hour Army qualification tests. The GPU-3 is expected to be essentially "inaudible" at a distance of 100 meters and to operate at 27 percent maximum thermal efficiency.

GM Research also has a 10-hp(s) Model 1036R engine which is used for research purposes and which is similar to the ground power engine.

The Allison Division worked on a Stirling-cycle solar-energy power plant for aerospace application from 1959 to 1963 under an Air Force contract. The engine was similar to the GM Research engines, and delivered 7 hp(s) with a 4.95-cu in.-displacement single cylinder at 30 percent thermal efficiency. The latest engine in this development was designated the PD-67.

The Electromotive Division has built 1-, 2-, and 4-cylinder units of the 8015 engine which develops 95 hp(s) per cylinder at 1500 rpm. A 4-cylinder, 360-hp(s) version in a generator set was tested by the U. S. Navy. Presumably, this engine could also be used for marine propulsion.

The Philips Company also has a number of Stirling-engine configurations. The single-cylinder Model 3015, developing 40 hp(s) has been subjected to sound emission tests in this country by the U. S. Navy. Philips is also developing a 4-cylinder, 236-cc displacement-per-cylinder marine unit to deliver 120 hp(s) at 3000 rpm.

TABLE 13. TYPICAL STIRLING-ENGINE DATA

Developer	Model No.	No. of Cylinders	Rating, hp(s) at rpm	Max. Brake Thermal Efficiency, percent	Envelope Volume[(a)], ft^3	Specific Volume[(a)], ft^3/hp(s)	Weight[(a)], lb	Specific Weight[(a)], lb/hp(s)
GM Research	GPU-2	One	7.5 at 3600	23	3.04	0.40	90[(b)]	12.0
GM Research	GPU-3	One	10.0 at 3000	27	4.89	0.49	188	18.8
Philips	3015	One	40.0 at 2500	39	6.40	0.16	550	13.8
GM Electromotive	8015	Four	380 at 1500	30	130.00	0.34	5000	13.2
Philips	Marine	Four	120 at 3000	40 (calc.)	23.20	0.19	725	6.0
GM Allison	PD-67	One	7 at 3000	30	3.50[(c)]	--	186[(c)]	--

(a) Reciprocating unit only, including combustor and preheater. Excludes fuel pump, water pump, combustion air blower, motor, radiator and fan assembly, working-fluid reservoir, starter motor, ignition system, and fuel control.

(b) Excludes flywheel also.

(c) Includes 4 hp(e) generator.

Current Experimental Automotive Units

The only known automotive application of the Stirling engine was built and tested by the General Motors Research Laboratories. The vehicle was a modified "Corvair" with a 30-hp(s) modified Series 3015 Stirling power plant using a thermal-energy storage unit mounted in the front luggage compartment. The thermal-storage medium was an aluminum oxide matrix. The vehicle, called the "Calvair", is said to have a 60-mile range. The need for a large heat-rejection capability is evident from the large radiator which takes up most of the rear of this "fastback" vehicle.

The coolant temperature on current GM Stirling engines is limited to about 160 F to obtain good life on the power piston rings and seals. In contrast, coolant temperatures on modern automobiles and trucks range between 170 F and 240 F. The Stirling engine rejects about 50 to 60 percent of the heat released by combustion to the coolant, while a modern IC engine rejects on the order of 25 percent through the radiator. The combination of these factors require a Stirling-engine vehicle to have a radiator with approximately 4 times the size of a conventional IC engine radiator.

In general, the transmission requirements for a Stirling-engine vehicle can be similar to or slightly simpler than those of IC engine vehicles. This is because the Stirling engine can be designed to have a torque characteristic that rises somewhat more than that of IC engines as engine speed slows down.

General Motors is reported to have plans to place Stirling-cycle power plants in several city buses for evaluation. These engines will be designed to produce performance equivalent to that of currently installed diesel engines.

Emission Characteristics

As discussed in the section External Combustors, very low mass-emission levels of CO and HC have been reported for Stirling engines (i. e., less than 1 percent of those of comparable uncontrolled-emission gasoline engines). While reported NO_x emissions are lower than for comparable uncontrolled-emission gasoline engines, considerable variation exists between engines, and reduction of NO_x emissions should be possible through directed research.

"1980" Physical, Performance, and Cost Characteristics

Specific Weight. In the range of 180 hp(s) to 16 hp(s), a specific weight for the reciprocating assembly of 5 to 6 lb/hp(s) should be obtainable by 1980. This includes combustor and preheater, but excludes the radiator, flywheel, and other accessories. The Philips' marine engine in Table 13 approaches this weight. Assuming a typical radiator and fan weight of 40 lb for a 150-hp(s) conventional automotive gasoline engine, and assuming that a radiator 4 times as large is required for automotive applications of the Stirling engine, a figure of about 1 lb/hp(s) should be added for the radiator. In addition, other accessories are estimated to weigh about 2 to 3 lb/hp(s). This puts the specific weight of a complete Stirling-engine power plant for vehicle application in the range of 8 to 10 lb/hp(s). The transmission weight would be about the same or slightly less than that of conventional gasoline-engine automobiles.

Specific Volume. Assuming advances in compactness, particularly with respect to component configuration and nesting to reduce maximum envelope dimensions, can be achieved, the specific volume of the basic engine, combustor, and preheater assembly may approach 0. 07 to 0. 10 ft^3/hp(s) over the 180 to 16-hp(s) range. Required radiator frontal area would be approximately 0. 06 ft^2/hp(s), and the volume required for radiator, fan, and other accessories would run an additional 0. 07 to 0. 10 ft^3/hp(s). This results in a projected Stirling-engine specific volume of approximately 0. 15 to 0. 20 ft^3/hp(s).

Efficiency. The most recent design studies indicate that larger sized Stirling engines are capable of operating with maximum brake thermal efficiencies as high as 40 percent. The particular engine for which this study was made, however, did not have a cooler fan. With design compromises to reduce cost, no increase is expected in this value by 1980. Smaller engines would have peak efficiencies between 30 and 40 percent. Figure 19 shows the maximum thermal efficiency that might be considered at part load. In general, the best part-load efficiency would be obtained at moderate engine speed with moderate-to-high working pressures, rather than at high speed and low working pressure.

Start-up Characteristics. Start-up time from the cold condition is determined by design variables. A start-up period of 10 to 15 seconds can be achieved; however, necessary design compromises may require a period of up to 30 seconds. Restart should be virtually instantaneous. As with conventional automotive engines, several minutes' driving would be required in cold weather to warm up crankcase and power-train lubricants before full power could be developed.

Power-Output Characteristics. The torque characteristic of Stirling engines droops, as speed in increased, somewhat more than that of gasoline or diesel engines, although the specific torque characteristic is a function of design parameters in any of these engines. However, the Stirling engine torque droops naturally as speed is increased owing to reduced effective heat-transfer capacity and increased aerodynamic-friction losses. Figure 20 shows a typical torque characteristic which rises about 20 percent as speed is reduced to 30 to 40 percent of maximum. Torque drops off sharply at lower speeds owing to leakage past the power piston. With the drooping torque curve, power drops off less rapidly as output speed is reduced. If desired, greater torque droop could be obtained artificially by a scheduled reduction in working pressure with speed. Of course, this would require a derating of a given displacement engine.

The maximum output speed of current Stirling engines tends to be lower than that of conventional automotive gasoline engines. The maximum continuous speed of a 10-hp(s) power plant would be about 3600 rpm, while a 200-hp(s) unit would run up to 1500 to 2500 rpm, depending upon the number of cylinders. Units designed specifically for automotive applications would probably have somewhat higher peak speeds. Idle speeds would probably be about the same or somewhat lower than that of conventional gasoline engines.

Throttle response would be excellent, perhaps equivalent to that of the diesel.

Power-Surge Capability. It would be possible for a Stirling engine to have a small power-surge capability in excess of the maximum steady-state power output, provided the reciprocating unit was built with the additional capacity. This power-surge capability has already been partially accounted for in that the radiator and fan have been sized with reference to present automotive radiators rather than with reference to current Stirling-engine practice which is based primarily on continuous rated power operation.

Specific Cost. Actual production cost figures have not been established; however, General Motors' cost studies indicate potential costs 15 to 20 percent higher than those for diesel engines. Some of the engine cost could be offset by the use of a simpler transmission; however, this saving in turn would be offset by the larger radiator required. Assuming that diesel engines for automobile application would cost twice as much as gasoline engines for this application* and that automobile gasoline engines cost \$2 to 3/hp(s), the Stirling power plant specific cost is estimated at \$5 to 7/hp(s).

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*Diesel engines for truck and bus application cost something like \$15 to 20/hp(s). However, if they were produced to meet the less severe automobile duty cycle and in the quantities required for automobile application, they could be expected to cost substantially less.

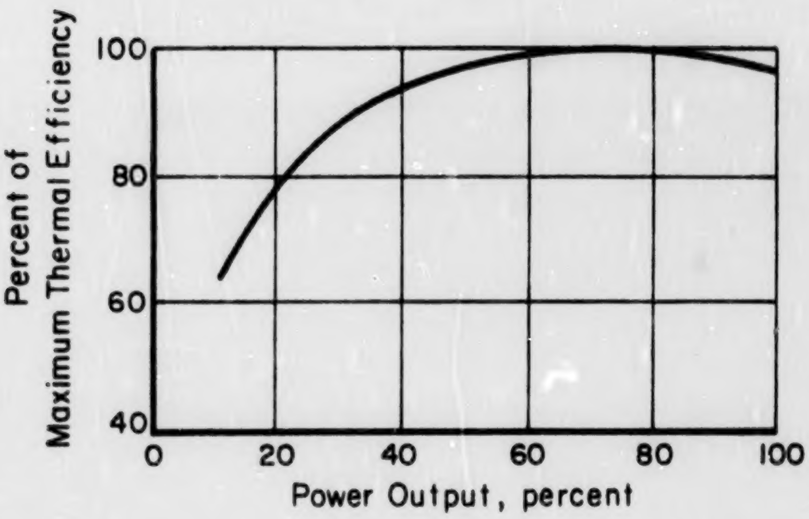

FIGURE 19. TYPICAL STIRLING-ENGINE PART-LOAD EFFICIENCY CHARACTERISTICS

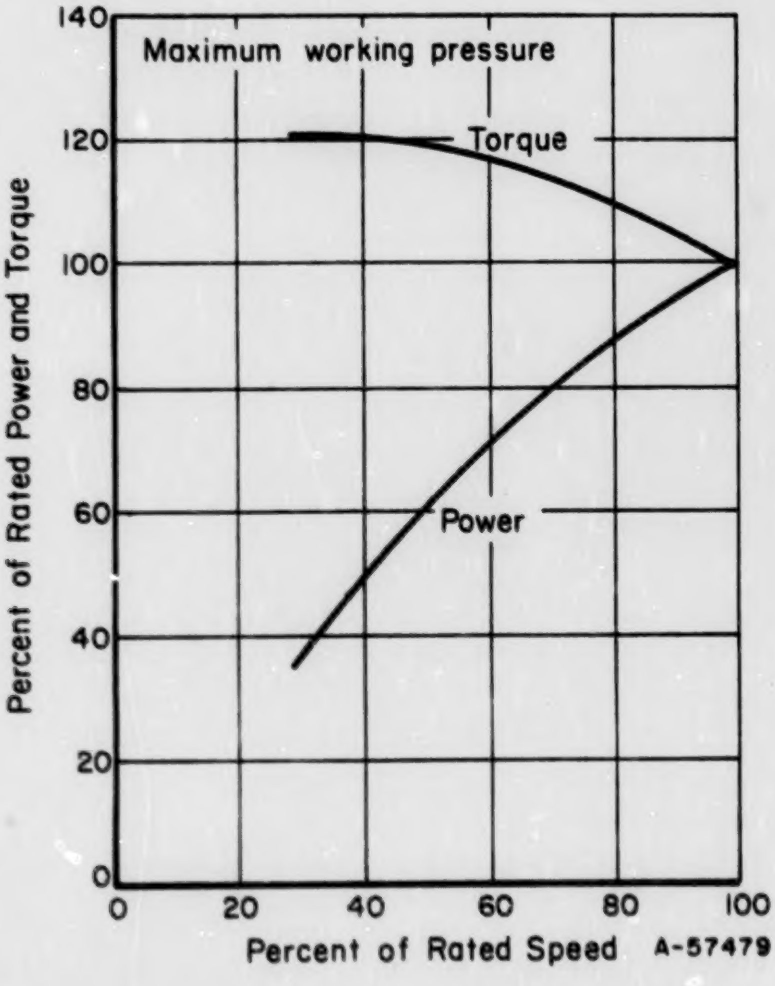

FIGURE 20. TYPICAL STIRLING-ENGINE TORQUE AND POWER CHARACTERISTICS

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DIRECT THERMAL-TO-ELECTRIC ENERGY CONVERTERS

MAGNETOHYDRODYNAMICS

General Description

Principles of MHD

Magnetohydrodynamic (MHD) power generation is based on the same principle as a conventional rotating-armature generator, that is, a conductor moving through a magnetic field causing an electric current to be induced in itself. The MHD generator, instead of using a solid metal conductor as the moving element, utilizes an ionized gas or a liquid metal flowing at high velocity through a magnetic field. A current is produced in the gas at right angles to the direction of flow, and electrodes placed in contact with the fluid carry the current to the load. Figure 21 illustrates the basic elements of an MHD generator.

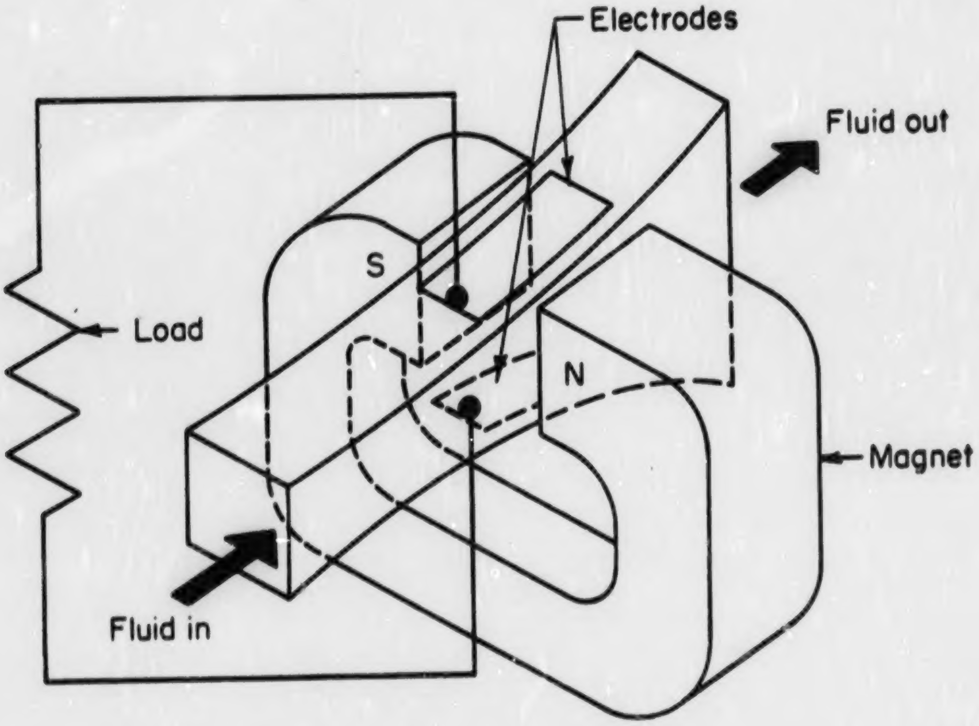

FIGURE 21. SCHEMATIC DRAWING OF MAGNETOHYDRODYNAMIC GENERATOR

The power produced by an MHD generator depends on the velocity of the fluid, its conductivity and the strength of the magnetic field. The absence of moving parts permits operating temperatures significantly higher than those possible in dynamic heat engines, thus providing a higher Carnot efficiency potential. However, high Carnot efficiencies cannot be attained in practice by simple MHD power system because the heat-sink or exhaust temperature must be relatively high to maintain the gases in a conducting state.

MHD Cycles and Requirements

There are three basic cycles that can be used with MHD power generation: the single open cycle, the open cycle with heat recovery, and the closed cycle. The simple open cycle involves a once-through pass of the working fluid with no recirculation and no attempt to make use of the heat remaining in the discharged gases. In the open cycle with heat recovery, the heat energy available in the discharge gases is recovered in a heat exchanger. This recovered heat may be utilized to heat the air for combustion or to generate steam for additional power.

Figure 22 is an example of an open-cycle MHD power system with heat recovery. In this example, the exhaust gases are first passed through a regenerator, where some of the available heat is transferred to the combustion air, and then through a boiler where steam is generated as part of a conventional Rankine-cycle, rotary-generator power system.

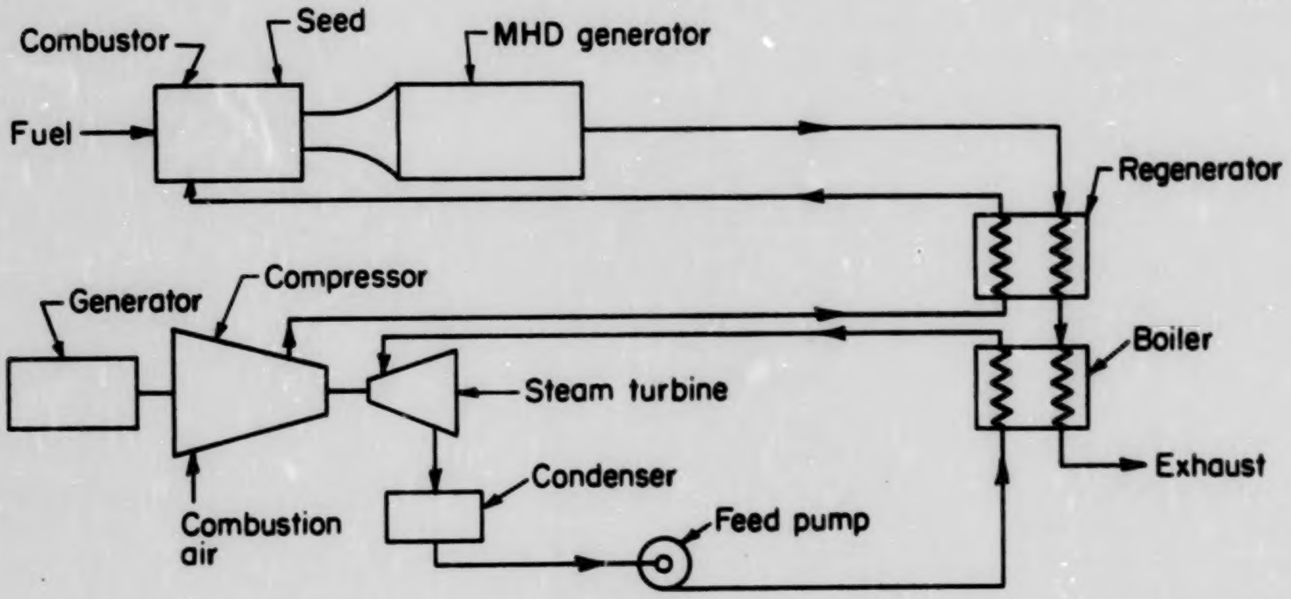

FIGURE 22. SCHEMATIC DIAGRAM OF OPEN-CYCLE MAGNETOHYDRODYNAMIC POWER SYSTEM WITH HEAT RECOVERY

The closed-cycle MHD generator involves complete recirculation of the working fluid. Figure 23 illustrates a closed-cycle, nuclear-powered, MHD-steam power system. The MHD working fluid passes through the nuclear reactor, through the MHD generator, through a boiler, through a compressor, and back to the nuclear reactor. Heat is transferred from the MHD working fluid in the boiler to generate steam for the Rankine-cycle, rotary-generator power system.

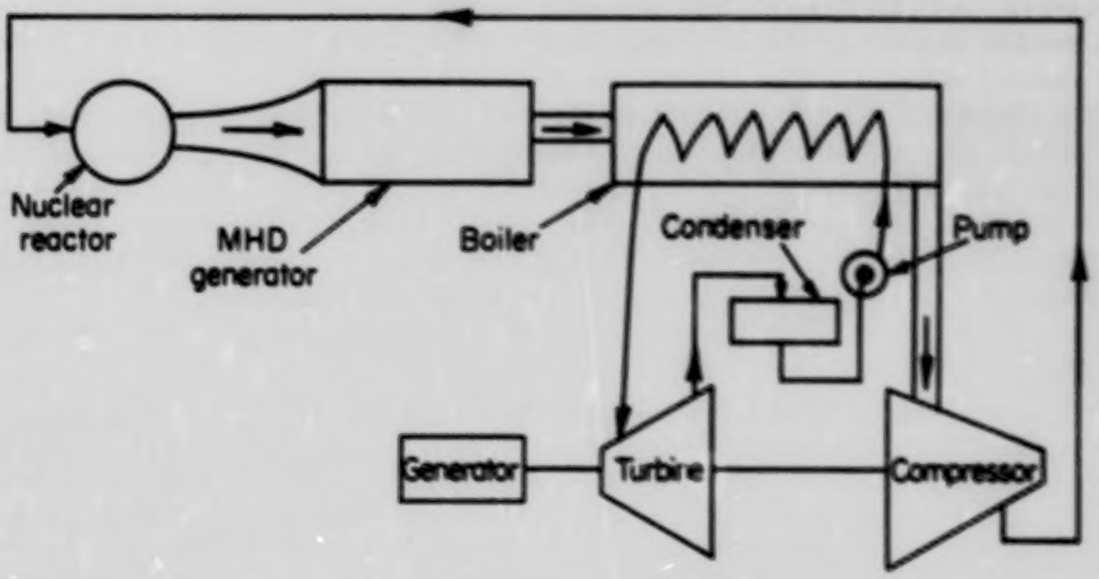

FIGURE 23. SCHEMATIC DIAGRAM OF CLOSED-CYCLE, NUCLEAR-POWERED, MAGNETOHYDRODYNAMIC-STEAM POWER SYSTEM

When a gas is used as the working fluid in an MHD generator. It must be ionized in order to be conductive. Ionization of gases can be accomplished in a number of ways, two of which are thermally and magnetically. Thermal ionization has received the most attention in MHD development because it is more easily achieved and appears to be a more practical approach.

For successful thermal ionization, not only must the gases be at a high temperature, but small amounts of a metallic salt must also be added. The minimum temperature at which an MHD generator can be expected to produce useful power is in the range of 3000 to 4000 F. Temperatures as high as 5000 or 6000 F are required to achieve high conversion efficiencies. Alkali metals such as cesium, rubidium, and potassium have been found to be the best seed materials. Potassium is most commonly used because of its availability and relatively low cost. The optimum quantity of seed material is about 1 or 2 percent of the total mass of the working fluid. Greater percentages actually result in a reduction in the ionization potential of the gas. Most fuels when burned with the proper amount of air will produce flame temperatures on the order of 3500 F, which is marginal for an MHD generator. Increasing the percentage or richness of oxygen in the combustion air or preheating the combustion air may be essential in a practical MHD system.

Current and Projected State of the Art

Experimental Accomplishments

Serious MHD development work is being pursued by only a relatively few organizations. Among these are several universities such as Stanford University, the University of Illinois, and Sheffield University in England. Corporations involved in MHD work include: AVCO-Everett, Westinghouse, Atomics International, and General Electric.

As an example, AVCO-Everett has recently tested an MHD generator utilizing the exhaust from a rocket engine as the working fluid. This system produced 31,600 hp(e) for a period of about 1 minute. The magnet of this unit was uncooled; consequently, a longer run time would not be possible without danger of burning up or melting the system. During the run the gas flow was about 6900 pounds per minute and the seed, potassium hydroxide, represented about 1-1/2 percent by volume. The fuel for the rocket engine was ethyl alcohol plus oxygen. The generator portion of the experimental unit was on the order of 25 feet long and 8 by 8 feet in cross section. The field strength of the magnet was about 36,500 gauss, and 134,000 pounds of copper was used in the magnet. The temperature of the working gases entering the magnetic field was about 4950 F.

At Westinghouse an experimental MHD generator using fuel oil plus oxygen, with potassium as the seed material, developed about 13.4 hp(e) for 10 minutes. Atomics International is working on liquid-metal alternating-current MHD generators. They have succeeded in inducing 10 hp(e) output using NaK as the working fluid in a blowdown system. At General Electric, experiments are being carried out on a nonequilibrium MHD generator using seeded inert gases. This type of MHD generator eventually is expected to be appropriate for use with a gas-cooled nuclear reactor as the heat source because the MHD fluid temperatures can be much lower for a given level of conductivity. Work is also going on at a number of universities and at AVCO-Everett on the feasibility of using powdered coal as the fuel for an open-cycle, regenerative-type MHD system.

Development Problems

The principal problems facing further development of the MHD concepts are: the high working-fluid temperatures required, recovery of the seed material, erosion of the electrodes, corrosion by the seed material, recovery of the exhaust heat, and the field strength of the magnet.

The high temperature requirements pose a fuel and a combustion problem and also lead to almost an absolute necessity for making use of the heat energy in the exhaust gases. The combustion of normal hydrocarbon fuels with air does not lead to high enough temperatures to result in an efficient MHD generator. Preheating the combustion air leads to more practical energy outputs, and of course the substitution of oxygen for some or all of the combustion air will result in even better energy-conversion efficiencies. As was previously mentioned, the heat still available in the gases discharging from the MHD generator can be used effectively to preheat the combustion air and/or to generate steam in a Rankine-cycle power plant.

Because of its cost, recovery of the seed must be considered essential in a practical system. This poses many problems, particularly when powdered coal or Bunker C oil is used, since these fuels produce a considerable amount of ash which would interfere with successful and complete recovery of the seed.

Regarding erosion of the electrodes it has been suggested that consumable electrodes be used in a manner similar to the electrodes in an arc furnace. Whatever final electrode materials and configurations are developed, it is probable that, replacement of the electrodes will be a maintenance requirement in MHD generators.

The maximum potential magnetic-field strength of conventional magnets today may be on the order of 35,000 to 40,000 gauss. It is hoped that advances in the technology of so called superconducting magnets will soon make possible much higher magnetic-field strengths, say on the order of up to 100,000 gauss, and thus will hasten the development of practical MHD systems.

Corrosion by the seed material and erosion by the combustion gases, which in the case of powdered coal would contain coal slag, are serious problems both within the MHD generator itself and in any heat exchangers used to recover the exhaust heat. These problems are further complicated by the fact that the walls must be electrically insulating and must be able to withstand high temperatures.

Emission Characteristics

Unburned hydrocarbons and carbon monoxide in the exhaust gases from an MHD generator should be negligible. However, the appreciably higher temperature essential for MHD power generation would probably lead to emission of nitrogen oxides in quantities considerably greater than those in a normal combustion system. In open-cycle systems, traces of the seed material might be present.

"1980" Physical, Performance, and Cost Characteristics

Work on MHD power generation is in such an early state of development that most of the characteristics of interest to this study are not yet defined, particularly when considering automotive application. While a number of prototype commercial MHD generators should be in operation by 1980, a number of significant development problems, as discussed above, remain to be satisfactorily resolved.

Applicable Size Range. Most researchers in the field have concluded that the MHD generator is primarily suited to the production of large amounts of power. It has been estimated that the lower power limit for a useful MHD generator is about 1 megawatt [1340 hp(e)]. The principal reason for this minimum practical-power-level limitation is that all of the loss mechanisms, such as viscous drag on the walls, heat transfer to the walls, losses associated with the magnetic field coil, etc. are all items that are proportional to the surface area of the device, while the power output is proportional to the volume. Thus, as the size or volume of a given MHD generator is reduced, the surface-to-volume ratio increases, hence increasing the losses relative to the power output until a point is reached where the final net power output is not worth the trouble.

Efficiency. The potential thermal efficiency of an MHD system without heat recovery in the exhaust can probably never exceed 10 to 15 percent. However a combined MHD-steam power plant can probably be 5 to 10 percent more efficient than a steam power plant alone.

Specific Cost. It has been estimated that the capital cost of MHD power plants may be somewhere between $40 and $110/hp(e) in 10 or 15 years. This applies to multimegawatt [1000's of hp(e)] sizes and is comparable to costs predicted for central-station steam power plants in the same time period.

THERMOELECTRIC CONVERTERS

General Description

The physical basis of thermoelectric generation of power is the Seebeck effect, or generation of voltage when two dissimilar metals or semiconductors are joined together. In order for the voltage thus generated to be sensed externally, the junction of the two materials must be maintained at a temperature higher than the opposite ends of each segment of material. The output voltage of such a junction is proportional to the temperature difference between the hot and cold ends of the metals and to the difference between an intrinsic property known as the Seebeck coefficients of the two metals.

A second phenomenon of importance in thermoelectric-power generation is the Peltier effect, which is the pumping of heat into or out of a junction when a current passes through the junction. In a thermoelectric generator, the Peltier effect tends to lower the hot-junction temperature and raise the cold-junction temperature. The rate of heat flow into or out of a junction is proportional to the current, the algebraic difference of the Seebeck coefficients of the two materials, and the absolute temperature of the junction.

A third thermoelectric effect known as the Thomson effect occurs when the Seebeck coefficient of a thermocouple leg varies with temperature. Heat is generated in the leg at a rate dependent on current, temperature, and the rate of change of Seebeck coefficient with temperature. As a rule, the Thomson effect can be ignored if the mean Seebeck coefficient of the materials over the applicable temperature range is used.

Historical Development

The Seebeck effect has been extensively used in temperature measurement but has only been used for power generation since the late 1940s. Although the metals known to exhibit relatively high Seebeck voltages were low cost, the efficiencies of electrical power generators using these metals was 1 percent at best. Telkes(72) in a 1947 article which reviewed the state of thermoelectric-power generation at that time, showed that materials containing bismuth and antimony, tellurium compounds, and lead compounds could be used to achieve power-generation efficiencies of about 5 percent. During this same period, the gas-controls industry was conducting research on thermoelectric generators which could use the heat from gas burners and pilot lights to generate the electrical power required to operate blowers and automatic controls.(73)

The advent of semiconductor transistors and diodes with consequently increasing knowledge of semiconductor technology – control of resistivity, control of purity, methods of material preparation – coupled with requirements of the space program, led to heavy expenditures on research in all phases of thermoelectric power generation. From 1958 to 1962, the U. S. Navy Bureau of Ships sponsored many research programs aimed at finding new materials capable of high-temperature operation, and at improving the thermoelectric efficiency and general utility of known thermoelectric materials. Since 1962, the emphasis in thermoelectric-power-generation research has shifted from materials development to hardware development. Improved electrical contacts, improved burners, and stronger and more chemically stable thermocouples have served to improve reliability, reduce weight, and bring actual generator efficiencies closer to theoretically possible values.

Current and Projected State of the Art

There is no technical or natural limit to the power capacity of a thermoelectric generator. Power output is directly proportional to the number of thermocouples for given heat-source and heat-sink temperatures; there is no optimization of power or efficiency to be achieved by increasing or decreasing size. As a consequence, thermoelectric generators presently are used in applications where one or more of the following requirements exist:

(1) Power required is less than 0.4 to 0.7 hp(e).

(2) Silent operation is necessary.

(3) Frequent maintenance, predictable or unpredictable, is undesirable or impossible.

(4) Reliability and long life are required.

(5) Operation from any heat source is required.

Advanced Development Units and Applications

Thermoelectric generators have been built and operated satisfactorily in a variety of situations. They have been operated from many different heat sources, including butane, gasoline, kerosine, propane, JP-4, and fuel-oil burners; radioisotopes; and nuclear reactors. With the exception of one generator which produced over 6.7 hp(e), all of the generators have been in the subhorsepower range.

A brief description is given of several thermoelectric generators which have been built recently and which could be classified as advanced development models.

0.4 hp(e)-Marine Engineering Laboratory. This generator described by Neild,(74) contains a thermoelectric section, a fuel tank with capacity for 8 hours' operation, a liquid fuel burner, and a fan for cooling the thermocouple cold junctions. The burner uses diesel, JP4, or kerosine fuel.

The thermoelectric section consists of 30 modules mounted in a cylinder surrounding the flame. Each module contains eight lead telluride thermocouples. The thermocouples are mounted in a hermetically sealed enclosure to prevent oxidation. Cold-junction temperature is maintained at 350 F by forced convection cooling of aluminum fins that receive waste heat from the thermocouples. The thermoelectric section, including cooling fins, weighs 13.9 pounds. The hot junctions are kept at 1050 F. Although service-life data are limited, it appears that generators of this type have operated successfully in excess of 1000 hours.

Statistical data on this generator are:

Power Output	0.4 hp(e), 26.8 v, 11.2 amp
Weight Without Fuel	22.5 lb
Dimensions	11 x 18 x 17 in. (1.95 ft^3)
Efficiency	3.7 percent
Specific Weight	56 lb/hp(e)
Specific Volume	4.9 ft^3/hp(e)
Start-up Characteristics	8 minutes with wick preheat; 3 minutes with propane preheat

0.4 hp(e) Vehicle Mounted Power Supply, U. S. Army Electronic Command. This generator(75) is intended to be used to charge batteries for communications equipment. It is carried on a jeep or other vehicle and uses the vehicle fuel supply. The entire unit consists of thermoelectric converter, burner system, fuel system, cooling system, and power conditioning, electrical control, and monitoring circuitry.

The thermoelectric section is similar to the thermoelectric section of the 0.4 hp(e) MEL unit described above. Lead telluride thermocouples are mounted in a hermetically sealed cylindrical unit which surrounds the burner. Fins are used on both the hot side and the cold side to enhance heat transfer to and from the modules. The converter weighs 12.2 pounds.

The burner uses liquid fuel which is atomized for combustion by an ultrasonic atomizer. Primary air is supplied by a blower. Fuel is pumped in by means of an electric fuel pump through an input connection provided. A fuel metering valve is also provided.

Thermocouple cold-junction temperature is maintained by means of air driven by an electric fan over aluminum fins.

Power conditioning circuitry regulates the output voltage to 29 volts and an oscillator is used to drive the ultrasonic atomizer. Controls provide for start-up and run conditions and protect against circuit overload.

Statistical data are given below:

Power Output	0.4 hp(e), 28 v, 10.7 amp
Dimensions	13-1/4 in. in diameter x 23 in. high (1.83 ft^3)
Weight	25 lb
Efficiency	Not known
Specific Weight	62 lb/ hp(e)
Specific Volume	4.5 ft^3/ hp(e)
Start-up Time	5 minutes

0.75 hp(e) Vehicle Mounted Power Supply, USAMERDC. This unit differs from the unit described above only in power output, size, and weight. Data are as follows:(75)

Power Output	0.75 hp(e), 28 v, 20 amp
Dimensions	14-5/8 in. in diameter x 25 in. high (2.44 ft^3)
Weight	35 lb
Efficiency	3.5 to 4.0 percent at rated load point
Specific Weight	47.5 lb/hp(e)
Specific Volume	3.30 ft^3/hp(e)

Other Low-Power Generators. In addition to these generators, a number of others have been described in the literature in recent years. For several years the U. S. Army Electronics Command has been developing a silicon-germanium-based generator that uses a multifuel burner and generates 0.13 hp(e). Net efficiency of this system is 2.75 percent. Specific weight is 330 lb/hp.

General Instrument Corporation markets a line of thermoelectric generators which utilize flameless, catalytic burners.(76) The generators can use either butane or propane gas and are available in sizes up to 0.4 hp(e). The thermoelectric material used is bismuth telluride.

Plevyak(77) has described a 0.2 hp(e) propane-fired thermoelectric generator used to power remote microwave telephone equipment. Two classes of generators were field tested under adverse conditions. One was a lead telluride-based system which was heated by a propane flame burner. The other was a bismuth telluride-based system which was heated by a catalytic propane burner. A cost analysis was performed by Plevyak, assuming a 30-year life of the microwave equipment, and a useful life of 5 years for the thermoelectric generator. He compared costs per year for power generated by a thermoelectric generator with cost per year for power from batteries and from engine-generator sets. In the range up to 0.4 hp(e) the thermoelectric generator provided the lowest cost power. Above 0.4 hp(e), the engine-generator sets were cheapest.

40 hp(e) Generator Feasibility Study

All of the generators described thus far have been in the fractional horsepower range. As has been indicated, situations calling for larger amounts of power have been better served by engine-generator sets. The principal advantages which thermoelectric generators can claim over the engine-generator set, silence and maintenance-free operation, have not been sufficient to outweigh the lower specific weight and lower cost of the engine-generator set. An exception to this generalization was an undertaking of the U.S. Army Transportation Research Command(78) to determine the feasibility of a 40 hp(e) thermoelectric generator to be used to power a silent boat driven by an electric motor. A thermocouple using silicon-germanium alloy for use in such a generator was designed and life tested. Thermocouples were built which demonstrated an efficiency of 10 percent (thermocouple efficiency only). The couples were run for several hundred hours in air, with hot-junction temperatures as high as 1920 F, with no degradation in output power. Thermocouples were also subjected to shock and vibration tests and did not show signs of cracks or increases in electrical resistance at the conclusion of the tests.

Following the successful silicon-germanium thermocouple development, the design of a 40 hp(e) system was undertaken to determine the size, weight, and general characteristics of such a system.

The final generator design called for a system of 42 self-contained subassemblies. Each subassembly contained four thermocouple modules of 81 thermocouples each surrounding a heat source which radiates energy at 2360 F. The complete generator was estimated to weigh 1000 pounds, with dimensions of 31 x 31 x 20 inches, or 11 cubic feet. The calculated overall efficiency was 6.5 percent. These figures imply a specific weight of 25 lb/hp(e) and a specific volume of 0.3 ft^3/hp(e).

The generator described here was not built. Although the contractor (RCA) considered such a generator feasible, he believed that some difficulty might be encountered owing to limitations of construction materials at the design-point operating temperatures. Nevertheless, these difficulties could probably be overcome in the course of a development program.

Probable Areas of Improvement

Future improvement in the efficiency, cost, and specific weight of thermoelectric generators could be achieved through: discovery of new alloys or compounds with increased figure of merit; improvement of existing materials; or improved construction techniques. The figure of merit, which is commonly designated as Z, is a measure of the usefulness of a material for thermoelectric application.

It is impossible to predict the discovery of new materials. A large amount of searching for new thermoelectric materials was done in the late 1950s and early 1960s. A great deal of information was gained on many materials. Criteria to serve as guidelines in investigating materials were formulated. However, the materials of maximum figure of merit at the present time, with the exception of germanium-silicon alloy, are simply modifications of materials which were known over 10 years ago. There is no known theoretical limit to the figure of merit, but it has been observed that the "ZT" product, or dimensionless figure of merit, never exceeds a value of 1.0 or 1.1. Very little research on thermoelectric materials is now being done. The advent of thermionics for use at temperatures above 2000 F has reduced the impetus for finding thermoelectric materials to operate in that temperature range. For these reasons, it would be unwise to hold out the hope for new materials showing significant improvements in the intermediate future.

Improvements in existing materials are likewise difficult to predict, particularly as far as the figure of merit is concerned. The technology of producing materials should steadily improve, and result in materials of more predictable properties and more stable properties than are now available. One might also expect improvement in the mechanical properties. The brittleness and low tensile strength of lead telluride are areas where improvement could be expected. Reductions in cost should follow increased production.

It appears that the best possibilities for thermoelectric-generator improvement are in the areas of design and construction techniques. The table below lists the particular aspects of device construction where improvement is needed, what the manifestation of this improvement should be, and what the prospects are for the particular improvement.

TABLE 14. AREAS WHERE IMPROVEMENT IS NEEDED IN THERMOELECTRIC-GENERATOR CONSTRUCTION

Area of Improvement	Effect of Improvement	Prospects
Reduce contact resistance	Increased efficiency; permitted shorter legs and higher power-to-weight ratio; reduced cost; improved reliability	Probably a slow, continuous improvement
Improved thermal insulation around thermocouples between hot and cold sides	Increased efficiency; permitted shorter legs with higher power-to-weight ratio and lower cost	Unknown
Use of cascaded thermocouples or segmented legs	Increased efficiency	Very good
Provide means to allow for thermal expansion mismatches at interfaces	Improved reliability	Good

Current Programs Aimed at Technological Improvement

Several current research programs which may provide insight into the future of thermoelectric generators are discussed in the following paragraphs.

Atomics International, Canoga Park, California, is working to improve efficiency by developing a cascaded thermoelectric module. A cascaded module is one in which two thermocouples are in series thermally so that heat rejected by the cold junction of the first-stage thermocouple is absorbed by the hot junction of the second-stage thermocouple. The first-stage thermocouple is made of materials which exhibit their maximum figure of merit in the temperature range of the first stage. The second-stage thermocouples are made of different materials, which exhibit their maximum figure of merit in the lower temperature range typical of the second stage. A cascaded generator can be operated effectively over a larger temperature difference than that possible with only one stage. A principal difficulty in cascading is in providing good heat transfer from the first stage to the second stage while providing electrical isolation.

The Atomics International work is in the early stages at this time.(79) Individual modules have been built and tested, and a design analysis has been carried out. The design objective is to achieve 10 to 11 percent module-only efficiency and a specific weight of 18.5 lb/hp(e) for the converter, exclusive of heat source and sink provisions. The module uses a silicon-germanium thermocouple stage operating between 1840 F and 1100 F and a lead telluride stage between 1000 F and 400 F. Maximum theoretical module efficiency for such a unit is approximately 14 percent.

Assuming that the thermoelectric modules constitute one-half the weight of the entire generator, a system weight of approximately 36 lb/hp(e) is projected for a system using these cascaded modules.

Another thermoelectric program(80) which could have results significant to this study is the TURPS (Terrestrial Unattended Reactor Power Supply) Program being carried out for the Air Force by the Martin Company. The heat source in the TURPS system is to be a nuclear reactor, but the thermoelectric section design and performance would be applicable to a hydrocarbon-fueled system.

The design goal of the TURPS program is to achieve 134 hp(e) based upon a thermoelectric converter with power-to-weight ratio of 10 lb/hp(e) for the converter portion alone. Converter-only efficiency has not been specified, but efficiency of individual couples has been 9 percent.

The TURPS system will probably use thermocouples having lead telluride n-type legs and lead-tin telluride p-type legs. The legs are designed with very high cross-sectional area-to-length ratios. Thermocouples under test have used legs 1.25 inches in diameter and 0.10 inch thick. The thermocouple legs are formed by pressing and sintering lead telluride powder in a very thin metallic jacket. The jacket provides strength and protection from contamination.

This program is in its beginning stages at this time. Thermocouples using different jacket materials have been designed and life tested, and the theoretical analysis of thermocouple performance has been done. The p-type materials showed severe degradation on life test. The problems of jacket material, pressing procedure, and overall element design have not yet been resolved.

The approach to thermocouple design used in this program, known as the high-power-density or HPD approach, is used to achieve higher power-to-weight ratios for a given material in a given temperature range. In principle, this is feasible.(81) However, the approach has several technical limitations. High power density implies a high thermal flux in the material, with resultant high thermal stresses. Lead telluride is quite brittle, and cracking due to thermal stresses has always been a problem. Furthermore, as thermocouple legs are made shorter and shorter, the effect of electrical contact resistance becomes greater, and thermocouple efficiency decreases. It appears that encapsulation might provide part or all of the answer to the thermal stress problem, but the problem of contact resistance remains. Presumably, development work will provide improved bonding techniques and result in lower contact resistances in the future.

Finally, a program being conducted at Battelle – Columbus which may result in higher efficiency thermoelectric generators is aimed at developing segmented thermocouple legs.(82) Segmenting differs from cascading in that there is only one generator stage in segmenting, as opposed to two in cascading. A segmented thermocouple leg is made up of sections or segments of different material. Each segment is located at the point in the leg where the temperature is most favorable for the material of which the segment is made.

The theoretical efficiency of a segmented silicon germanium-lead telluride thermocouple operating between 1800 F and 100 F has been calculated to be 15 percent for the thermocouple module alone. Such a figure has not been achieved in practice, however. The principal problems are related to achieving a suitably strong, low-resistance bond between the lead telluride and germanium-silicon. The problems are complicated by the thermal-expansion mismatch between the materials. Two approaches are used to solve this problem. One is to sandwich intermediate materials between the thermoelectric materials to take up the expansion stresses. However, the choice of intermediate materials and the method used to bond them to the thermoelectric materials is complicated by the sensitivity of the thermoelectric material to high temperatures and contamination. The second is to use a pressure contact between the lead telluride and germanium-silicon segments. Such contacts are relatively high in electrical resistance, however.

"1980" Physical, Performance, and Cost Characteristics

In estimating the potential performance of thermoelectric systems, the following are assumed:

(1) No improvement in figure of merit or temperature capability of existing materials will occur, nor will new materials of higher figure of merit be found.

(2) The high power density and cascade concepts will be successful and embodied in one generator.

(3) Burner efficiency will be increased to 80 percent.

(4) Approximately 10 percent of generated power will be used to drive blowers, pumps, and power conditioning systems.

(5) One-half of the total system weight will comprise the thermocouples, and one-half will comprise the burner, cooling system, and auxiliary equipment.

Specific Weight and Efficiency. At present, a 0.75 hp(e), 4 percent efficient system with a specific weight of 47.5 lb/hp(e) has been built. A 40 hp(e), 6.5 percent efficient system with specific weight of 25 lb/hp(e) is regarded as several years in the future. Cascaded modules designed for 11 percent efficiency and a specific weight of 18.5 lb/hp(e), module only, have been built, but not a complete system using cascaded modules. The potential efficiency of a cascaded or segmented module alone is 14 to 15 percent. High-power-density modules, 9 percent efficient and weighing 10 lb/hp(e), have been demonstrated.

According to the previously listed assumptions, a cascaded system, if built at the present time, would weigh 36 lb/hp(e) and would be 7.7 percent efficient; the high-power density system, if built at the present time, would weigh 20 lb/hp(e) and be 6.3 percent efficient. The potential efficiency of a cascaded or segmented system would be 0.7 (burner plus auxiliary losses) x 0.15, or approximately 10 percent. If such a system were built with the high-power-density design, the weight might be

$$\left(20 \frac{\text{lb}}{\text{hp(e)}} \right) \times \left(\frac{6.3}{10.0} \right) = 13\ \text{lb/hp(e)} \quad .$$

These figures represent the highest system efficiency and the lowest specific weight that can be envisioned at this time. However, the possibility of achieving this value by 1980 is reasonable.

Specific Cost. Cost of thermoelectric-generator systems in the high-power ranges is difficult to estimate. Costs of present generator systems are in the neighborhood of $11,200/hp(e). This cost is expected to decrease to approximately $3,000/hp(e) which is in the same ratio as the weight per hp(e). While building units at a rate of several hundred thousand per year would result in a drastic cost reduction, it is believed that the cost would still remain extremely high for automotive application.

THERMIONIC CONVERTERS

General Description

Thermionic conversion of heat to electricity is based upon application of the Edison effect, that is, the evaporation of electrons from hot bodies. As thermal energy is added to the body (emitter) additional energy is imparted to its electrons until some of them evantually are able to surmount the potential barrier at the surface. If a second electrode (collector) is placed near the emitter and the two electrodes are connected through an external circuit, some of the emitted electrons will possess sufficient energy to traverse the space between the electrodes, and (providing the potential barrier at the surface of the collector is smaller than that at the emitter) do useful work in the external circuit. The number of electrons (current) emitted by a heated body increases exponentially as the temperature of the emitting body increases. Thus, the conversion efficiency should also increase rapidly as the emitter temperature is raised.

This would be true except for the fact that the electrons in the space surrounding the emitter eventually begin to exert a strong repulsive force, owing to their similar electrical charges, on additional electrons which are attempting to leave the emitter. This layer of electrons which surrounds the emitter is normally called the space-charge cloud or just the space charge. In order to provide the high current densities necessary for reasonable efficiencies, some means of space-charge compensation must be provided.

It is possible to reduce the spacing between the emitter and collector to a small enough value so that space charge has very little effect upon diode performance. The spacings required, however, are extremely small, less than 0.001 inch, and the mechanical problems involved in such a scheme have led to abandonment of efforts to produce practical, close-space converters.

The alternative method of space-charge compensation that has been almost universally adopted is the introduction of positive ions into the interelectrode space. The usual source of ions is cesium vapor, which can be ionized by contact with the hot emitter.

For a given diode spacing and cathode temperature there is an optimum cesium vapor pressure. The performance of the converter is significantly degraded if the operating conditions depart appreciably from optimum.

Current and Projected State of the Art

Major characteristics of thermionic conversion devices which are usually considered advantageous are:

(1) There are no moving parts.

(2) Operation at temperatures up to 4000 F is possible.

(3) Waste heat can be rejected at relatively high temperature, i.e., up to about 1300 F.

(4) The power-to-weight ratio is relatively high.

The characteristics listed above make thermionic conversion highly attractive for potential long-duration space missions, especially since nuclear energy appears to be ideally suited to provide the thermal energy and there do not appear to be any major materials-compatibility problems. For this reason a major portion of the experimental research in thermionic conversion is directed toward developing nuclear-powered converters for space applications. Some work has been done on solar-powered converters, but the effort in this area has not been as great. Finally, work is being done to develop fossil-fuel-fired converters, primarily for military applications where portability or silent operation is the objective. The status of fossil-fuel-fired converters will be examined since this is the most appropriate to automotive application.

Current Experimental Units

As one might suspect, the major problem encountered in adapting thermionic converters to fossil-fuel-fired heating is prevention of oxidation of the hot metallic parts. Three organizations have been conducting research on this problem in the United States. Thermo Electron Engineering Corporation(83, 84, 85) (TEECO), Waltham, Massachusetts, and Consolidated Controls Corporation(86) (CCC), Bethel, Connecticut, have been developing silicon carbide flame barriers while Radio Corporation of America(87) (RCA) at Lancaster, Pennsylvania, has been working on a heat-pipe concept which uses an alumina tube exposed to the flame.

It is believed that units designed by TEECO are the most pertinent to this study. Present capabilities and anticipated improvements for the TEECO fossil-fuel-fired thermionic conversion system are enumerated in the discussion which follows.

A 6.7 hp(e) flame-heated power supply was designed at TEECO about a year ago. To our knowledge, this large unit has not been built, but individual modular diodes which would be used in it have been built and tested in 0.4 hp(e) experimental units. As designed, the high-temperature burner would heat the diode emitters to 2750 F and would be capable of converting 50 percent of the available heat of combustion of the fuel to heat for the emitter. Energy conversion is accomplished in 0.13 hp(e) modular diodes connected in series. One great disadvantage of thermionic converters is that they are low-voltage high-current devices with a typical output voltage of 0.5 to 0.7 volt. Thus, in the 6.7 hp(e) generator designed by TEECO, 48 diodes are connected in series and produce an output voltage of 24 volts. Provisions have to be made for failure of an individual diode in the open-circuit mode to prevent complete loss of power. The conversion efficiency of the diode itself is calculated to be 10 percent.

Data taken from the literature on theTEECO 6.7 hp(e) unit are:

Specific Weight	20 lb/hp(e) including batteries to power blowers and fuel pump during start-up, but not including power conditioning equipment, if required
Specific Volume	0.45 ft^3/hp(e)
Cost	Estimated by TEECO to be about $30 or less for each 0.13 hp(e) modular converter [i.e., 225 $/hp(e)], if mass produced. This cost is for the thermionic module only and does not include the cost of blowers, combustor, heat exchanger, etc.
Efficiency	As designed, about 5 percentof thermal content of fuel is converted to usable electricity.
Power-Output Characteristics	Each diode will provide about 0.4-volt output. Series – parallel connections are required to provide higher voltages.
Start-Up-Time	The best start-up times so far achieved with smaller units are just under 10 minutes.

Variations in power level can have extremely detrimental, sometimes catastrophic effects on thermionic converters, so that provisions for handling load fluctuations would have to be engineered into the system.

"1980" Physical, Performance, and Cost Characteristics

Continued research and development with fossil-fuel-fired converters will probably result in increasing the conversion efficiency of the diode itself to about 15 percent and in a doubling of system efficiency to about 10 percent by 1980. The heavy currents from such devices will probably limit the specific power per individual diode to about its present level in order to minimize lead losses.

In principle, any number of 0.13 hp(e) therimonic diodes can be stacked to build a power source of any desired power level. Maximum efficiency of the system requires that all diodes operate at very near their optimized temperature, however, so appropriate combustor and air preheater designs would have to be developed.

Assuming a doubling of system efficiency by 1980, system specific weight, size, and cost will be at least halved. Thus in 1980 it will probably be possible to build a system weighing about 10 lb/hp(e), having a bulk of about 0.2 ft^3/hp(e), and costing about $100/hp(e) for the thermionic module alone. It is unlikely that instant start-up can ever be achieved, but the lag may be reduced to only 3 to 4 minutes.

THERMOPHOTOVOLTAIC CONVERTERS

Thermophotovoltaic generation of electrical power (TPV) is a relatively new and undeveloped technique. As far as is known, no truly practical devices of this type have been built. Several low-power laboratory models have been built(88, 89) and a 4.0 hp(e) system was completely designed and analyzed, but was not built. Research on photocells for use in TPV converters is being conducted at MIT(91) and Delco.(92)

The problems standing in the way of development of practical systems that are competitive with other types of energy-conversion systems are known, and approaches to the solutions of these problems are under consideration. However, until more performance data are available, it is difficult to assess the potential of this type of converter for powering an urban vehicle.

General Description

The processes by which thermal energy is converted to electrical energy in a TPV converter are the following. The stored energy of a heat source is used to heat a body usually a cylindrical rod of silicon carbide, or a mantle, which then emits electromagnetic radiation. The emitted radiation impinges on a photovoltaic converter. That portion of the radiation which falls in a suitable wavelength range, to be described later, is converted into electricity. The remainder is either reflected back to the emitter and absorbed, helping to maintain the emitter temperature at a desired level, or is lost through scattering or absorption by passive components of the system.

A typical configuration is to have the photovoltaic cells mounted on the walls of a cylinder surrounding the emitter rod. Another configuration places the emitter at the focus of a parabola, with the photovoltaic cells mounted on a flat screen normal to the axis of the parabola.

The emitter is either a cylindrical rod of a refractory material such as silicon carbide, or a radiating screen such as the Auer-Welsbach mantle, which is a mesh screen of thorium oxide with 0.8 percent cerium oxide added to give strong spectral emittance in the visible wavelength band. The emitter is usually heated to about 2600 F. The requirements for the emitter are that it be capable of operating at high temperature for long periods and that it have a high radiant emissivity in the wavelength range of interest.

The photovoltaic converters are similar in construction and identical in their principle of operation to the solar cells used to provide power for spacecraft and satellites. However, because of the spectral characteristics of the energy emitted by a 2600 F source, germanium rather than silicon is best suited for TPV.

All photovoltaic cells are made of semiconductors which are characterized by an energy gap, E_g, which is defined as the minimum energy required to raise an electron to a "free" state. Only that radiation of wavelength, λ, such that $\lambda \leq \frac{1.24 \text{ micron-electron volts}}{E_g}$, where E_g is in electron volts, can be converted to electrical power in a photovoltaic cell of energy gap E_g. Furthermore, when radiation of energy greater than E_g is absorbed by the photovoltaic cell, the electrical-energy output cannot be more than E_g. Thus, the radiation of energy less than E_g is not converted at all, and only a fraction of the radiant energy greater than E_g is converted to electrical energy. Finally, part of the radiation striking the cell is reflected rather than converted. Reflection losses can be reduced by use of antireflection coatings, and by arranging the cells in such a way that reflected radiation returns to the source.

Current and Projected State of the Art

The photovoltaic cells must be made from semiconductors of very high purity and crystalline quality. The only materials from which reasonably good photocells have been made, and their energy gaps, are the following:

	Energy Gap, ev	λ for Energy Gap, microns
Germanium	0.66	1.88
Silicon	1.09	1.14
Cadmium telluride	1.44	0.86
Cadmium sulfide	2.38	0.52
Gallium arsenide	1.41	0.88
Indium arsenide	0.36	3.45

Of these materials, only germanium, silicon, and indium arsenide could be considered for converting the emitted radiation from a rod at 2600 F to electrical power. Germanium has been shown to be the best material for this application in view of its favorable energy gap and its cost, which is believed to be lowest of the three.

The efficiency of a TPV system can be expressed as:

$$\eta_{overall} = \eta_{cell} \times \eta_{sp} \times \eta_s \times \eta_{acc} \quad ,$$

where

$\eta_{overall}$ = overall efficiency, ratio of net electrical-energy output to thermal energy of fuel consummed

η_{cell} = cell efficiency, ratio of gross electrical energy to usable radiation energy

η_{sp} = spectral efficiency, ratio of usable to total radiation energy emitted by the source

η_s = source efficiency, ratio of total radiation energy emitted to thermal energy of fuel consummed, this accounts for all stack losses, and other system conduction and convection losses.

η_{acc} = accessary efficiency, ratio of net electrical energy to gross electrical energy; this accounts for losses due both to the electrical regulator required and to auxiliary drives.

The spectral efficiency η_{sp}, of systems using germanium and silicon photovoltaic cells and a 2600 F emitter was calculated as a function of black-body-emitter temperature by Kittl. He found that spectral efficiency for a germanium cell converter was approximately 25 percent if none of the nonusable radiation were returned to the source, and would increase monotonically to approximately 75 percent if all of the nonusable radiation were returned to the source. Spectral efficiency for a silicon cell system increased from approximately 5 percent with zero return to approximately 30 percent with 90 percent return, and thence to approximately 85 percent with 100 percent return of nonusable radiation to the source. This calculation shows the importance of returning nonusable radiation to the source.

Current Research Efforts

The following three approaches have been used in attempting to increase the return of unusable radiation to the source:

(1) Use of transparent cells provided with highly reflective coating on the back surface (nonabsorptive cells)

(2) Addition of chemical additives to the emitter to increase the emission in the usable energy range (selective emitters)

(3) Use of multilayer filters which, by means of optical interference, reflect unusable radiation and transmit usable radiation.

Nonabsorptive Cells. The first approach has had limited success for several reasons. Although highly pure germanium is transparent to radiation at energies lower than the energy gap, it is also highly electrically resistive. Selected impurities can be added to the germanium to reduce electrical resistivity and thereby increase cell efficiency, but the transparency decreases. Addition of metallic "fingers" to the front surface reduced cell resistance, but further reduced transparency. Cell efficiency as high as 22 percent has been reported, as opposed to 28 percent for absorptive cells. Haushalter, et al.(90), predicted a maximum product of cell and spectral efficiencies of 15.8 percent for a nonabsorptive system.

Selective Emitters. The second approach, use of a selective emitter, shows promise of success on the bases of results obtained by Kittl(89). The intent is "to find materials which have high spectral emissivity below the absorption edge wavelength in the region of maximum cell response and low emissivity in the near infrared beyond the absorption edge wavelengths". Calculations showed that metals such as tungsten, iridium, and platinum offer little chance of achieving the desired emission. A second class of materials investigated for this purpose, rare-earth oxides, was considered promising. A mantle of 70 percent thorium oxide and 30 percent erbium oxide showed a sharp emission peak at a wavelength of 1.5 microns.

An experimental test of TPV converters was conducted using several different rare-earth-oxide mantles. The mantles were open-mesh screens heated by a propane burner. An $\eta_{overall}/\eta_{aux}$ efficiency of 4.35 percent was achieved with a mantle of 10 percent erbium oxide and 90 percent theorium oxide. It was believed that higher efficiency could have been achieved with a closer mesh in the mantle.

The use of selective emitters is an interesting concept, and the results achieved by Kittl for erbium and thorium oxides were very promising. However, the ability of emitters of these materials to operate with stability at temperatures of 2600 F for long periods of time has yet to be demonstrated. The performance of these emitters depends upon the inclusion of trace impurities, which probably do not have vapor pressures the same as those of the host materials. Thus, the composition-dependent spectral characteristics of these emitters would be likely to change over a period of time. Further investigation of this point is required.

Absorptive Cells Used With Spectral Filters. A system consisting of an absorptive cell used in conjunction with filters which reflect nonusable energy to the emitter and transmit usable energy to the cell appears to be promising for achieving a high efficiency of conversion of radiant energy to electrical energy. The photovoltaic cells used in such a system are inherently more efficient than those used in a nonabsorptive system. Cells used by Kittl, for example, converted 28.3 percent of usable radiation impinging on them to electrical power, whereas nonabsorptive cells converted a maximum of 22 percent of the usable radiation to electrical power. There has been difficulty in obtaining filters of high spectral efficiency, although Haushalter et al. recently reported a filter with spectral efficiency of 70 percent. Haushalter calculated a maximum $\eta_{cell} \times \eta_{sp}$ efficiency of 22.5 percent for a system of absorptive cells and filters.

4.0 hp(e) System Design Study

Possibly the best indication of the current state of the art is an engineering study sponsored by USAMERDC, Ft. Belvoir, Virginia.(90) Six TPV systems were contemplated, all to operate in a 125 F ambient with the following power and weight:

0.4 hp, 35 pounds without fuel
0.4 hp, 35 pounds including 8 hours fuel
0.67 hp, 35 pounds without fuel
0.67 hp, 35 pounds including 8 hours fuel
4.0 hp, 100 pounds without fuel
4.0 hp, 150 pounds without fuel.

Comprehensive designs were carried out, and each part of each system was optimized for performance and weight.

It was concluded that the desired systems could not be built to operate with photovoltaic cells available today if ambient air at 125 F is to be used as the heat sink. Their data indicated that a specific weight of approximately 49 lb/hp(e) and an overall efficiency of 4.6 percent could be achieved on a 4.0 hp(e) system without fuel. If water at 80 F could be used for cooling the cells, specific weight would reduce to 37.5 lb/hp(e). These figures assumed cells with characteristics superior to those available now, but which could be developed in about 1 year.

The following efficiencies were calculated for the proposed 4.0 hp(e) system:

Heat of Combustion of fuel, hp(t)	87.0
Gross Electrical Power Generated, hp(e)	5.04
Net Electrical Power Generated, hp(e)	4.0
Source Efficiency, percent	48.1
Spectral X Cell Efficiency, percent	12.0
Gross Efficiency, percent	5.8
Net Efficiency, percent	4.6

"1980" Physical and Performance Characteristics

Three approaches to TPV conversion have been considered: use of nonabsorptive devices with reflective back surface, use of absorptive devices with selective emitters, and use of absorptive devices with filters. The absorptive cell-filter combination appears to be the most promising at the present time. This approach has a theoretical efficiency higher than that of the nonabsorptive-cell system. The selective emitter system is relatively untried, and even if it were successful, it seems unlikely to bring about efficiencies higher than those of the absorptive cell-filter system.

The point of departure for projecting future developments in TPV is the 4.0 hp(e) system designed by GM The product of spectral efficiency and cell efficiency was 12 percent. However, in the report it was stated that if available light filters could be successfully applied, this product could be increased to 22.5 percent. As discussed, Kittl reported a cell efficiency of 28.3 percent, and Haushalter reported a filter having a spectral efficiency of 70 percent. Therefore, at the present time, this product could possibly be as high as 19.8 percent.

In speculating on the potential specific weight and overall efficiency of a TPV system, it is reasonable to assume the following improvements:

(1) Increase source efficiency from 48 to 65 percent

(2) Increase accessary efficiency from 80 to 85 percent

(3) Increase product of cell efficiency and spectral efficiency from 12 percent to 22.5 percent.

With these three improvements, overall system efficiency should increase from 4.6 percent to 12.5 percent. If specific weight were inversely proportional to efficiency, the potential specific weight could be speculated to be reduced from 37.5 lb/hp(e) to 13 lb/hp(e). This specific weight is quite optimistic as the 37.5 lb/hp(e) specific weight was calculated for the water-cooled system.

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OTHER ENERGY STORAGE AND CONVERSION DEVICES

THERMAL-ENERGY STORAGE

General Description

A thermal-energy storage system is one in which thermal energy is stored in a material by means of some physical change in the material. Examples of physical changes in materials which can result in a release of stored thermal energy are: change from one crystalline form to another, condensation, solidification, and drop in temperature. All of these physical changes are reversible in the sense that heat supplied to the material will reverse the change of phase or temperature. Thus a thermal energy storage material can be cycled many times between reasonable upper and lower limits of thermal energy.

Current and Projected State of the Art

Thermal-energy storage has been and is being used in a number of ways. Hot-water storage has been used in industries and central power stations for steam generation during peak load periods. In these applications the water is not only heated but is also pressurized to increase its heat-storage capability. The interest in solar heating a number of years ago created a necessity for finding low-cost materials with the necessary qualifications for storing solar energy as thermal energy during the day and releasing it during the night. Thermal-energy storage has been investigated for use in the space program, particularly for earth satellite power systems depending on solar energy. Submarine propulsion systems, which must at times operate independently of atmospheric air, are another application of thermal-energy storage which has received some attention. Finally, at least one experimental thermal-energy storage Stirling-engine, vehicle propulsion system has been built.

Physical and Cost Characteristics – Thermal-Energy-Storage Materials

Materials absorb or release heat while undergoing one or more of the following physical changes: change from one crystalline form to another, liquefaction (melting) or solidification (freezing), sublimation, evaporation or condensation, the heat of solution and absorption, and a change of temperature in a single phase. Of these many possibilities the most important for practical application are the change from the solid to the liquid state and vice versa, and the change of temperature in a single phase. Both of these physical changes are completely reversible without affecting the chemical makeup of the material involved, occur in a useful temperature range for many common materials, and result in a fairly high magnitude of thermal-energy storage.

Figure 24 is a graphical illustration of the thermal-energy content of a typical substance versus its absolute temperature, showing three transitions as it changes from one crystalline form to another, followed by melting, and finally evaporation. The slopes of the curves and the heights of the vertical lines represent the quantities of heat energy that that can be stored in or released from a unit weight of the material. The temperature limits indicated by the vertical dashed lines represent the region in which this particular material would be useful in a thermal-energy storage system.

A good heat-storage material must have a high specific thermal-energy storage capacity over the operating temperature range of interest. It should have a low vapor pressure at the maximum operating temperature to avoid restrictions on the container configuration and to reduce container weight. High density is essential to reduce the bulk volume of the system. The material should be chemically stable at all temperatures so that the cyclic heating and cooling or change from liquid to solid will not result in deterioration of the material or its container. It must be noncorrosive to the container and any heat transfer surfaces in contact with it. It must be rechargable by some efficient method. The material should also be noncombustible and nontoxic for obvious reasons. Finally, it must be low cost and abundant.

Further restrictions are placed on the selection of a heat-storage material and its associated thermal-energy storage system by the particular requirements and operating characteristics of the application. These restrictions are dictated not only by the general application (such as, in this case, urban vehicles) but also and possibly more importantly by the device which is selected to convert the stored thermal energy to useful motive power. These requirements and operating characteristics include: temperature range; the relative importance of size, weight, efficiency, and cost; and the cycling frequency and rates of discharge and recharge.

The maximum temperature range within which thermal-energy storage systems could be expected to operate profitably is fairly easy to establish. The lower limit of this temperature range would depend on the available heat-sink temperature, which in the case of a vehicle is the temperature of the ambient air, and on the minimum acceptable energy-conversion efficiency. With ambient air as the heat sink, the lowest practical temperature limit would be about 500 F. The upper temperature limit is dictated largely by the problem of containment. If the container were metal, an upper temperature limit of 1600 or 1700 F would be acceptable. If the container were made of a refractory material, a much higher upper temperature limit could be allowed. As a compromise in the search and evaluation of potential heat-storage materials, an upper temperature of about 2800 F seems appropriate.

A fairly thorough screening of materials for heat capacities in the solid and liquid states, for heats of fusion, for usable temperature ranges has been carried out in recent years and reported in the literature(93, 94). Of the hundreds of materials studied, only a relatively few had high enough heat capacities or heats of fusion to be considered as competitive with other forms of energy for vehicular use. Even fewer actually met the other qualifications considered desirable for the application.

Table 15 presents pertinent data for five materials with heats of fusion over 0.12 hp(t)-hr/lb and with melting points within the usable temperature range. These compounds would be suitable in thermal-energy storage systems involving solidification of the material, to release the stored heat. Lithium hydride has the highest heat of fusion per pound of any substance known. Unfortunately it is also a very expensive

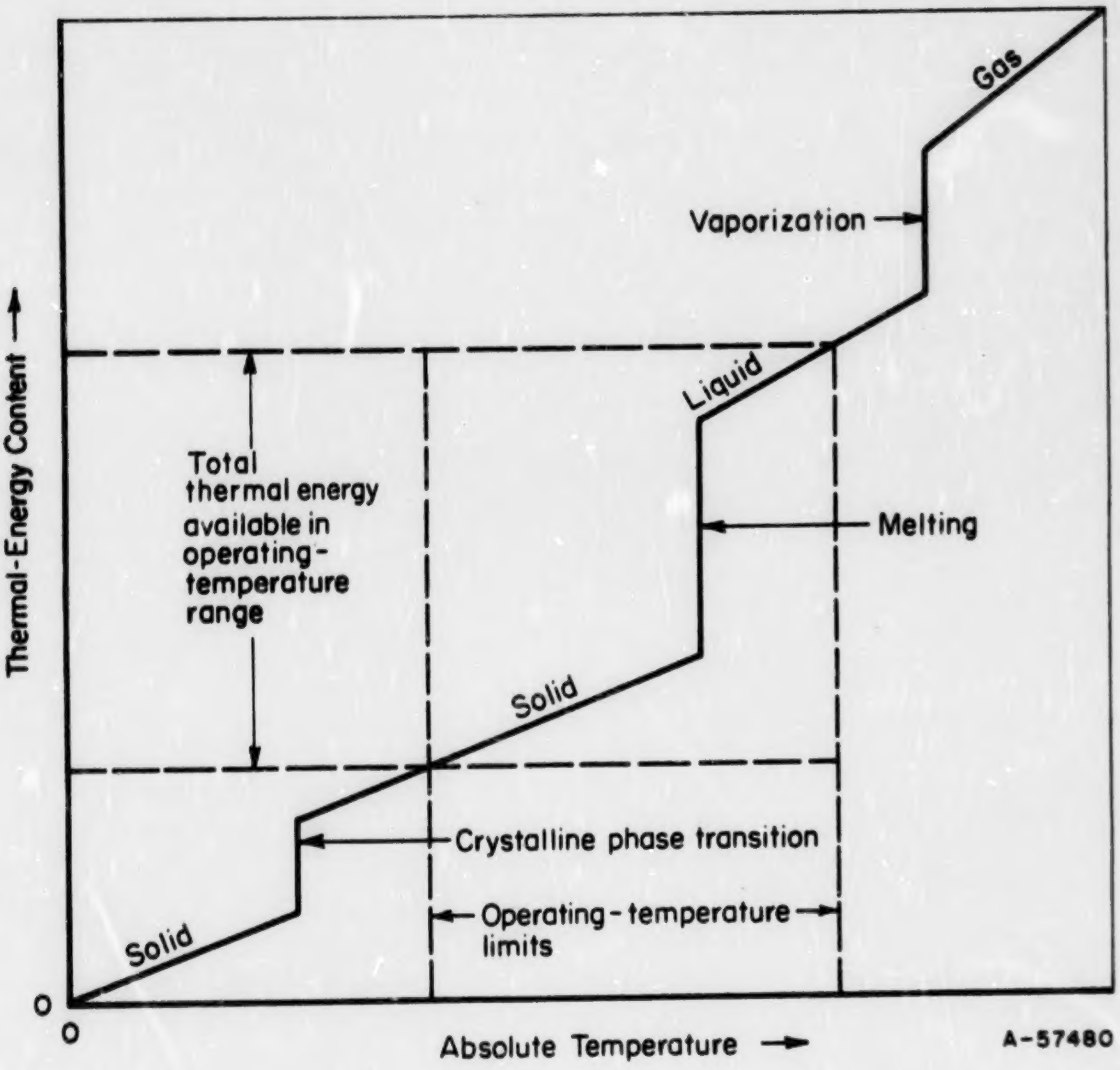

Thermal-Energy Content →
Gas
Vaporization
Liquid
Total thermal energy available in operating-temperature range
Melting
Solid
Crystalline phase transition
Operating-temperature limits
Solid
0
0
Absolute Temperature →
A-57480

FIGURE 24. THERMAL-ENERGY CONTENT-TEMPERATURE DIAGRAM FOR TYPICAL THERMAL-ENERGY STORAGE MATERIAL

material and has a low density which would result in a larger volume for a given heat capacity. Another potential disadvantage of lithium hydride is that it is flammable and will react violently with moisture in the presence of air.

Sodium fluoride and lithium fluoride appear to be attractive compounds for vehicle application. Sodium fluoride has the lowest cost in dollars per unit energy storage capacity, but its melting temperature may be a little high for metal containers. Lithium fluoride has the highest heat of fusion per cubic foot and is second only to lithium hydride in heat of fusion per pound. It also has a very favorable melting temperature. These characteristics of lithium fluoride would probably result in its being the best choice of the molten materials for vehicular application.

Table 16 presents pertinent data for a number of metallic oxides which have high specific heats in the temperature range of interest in this study and melting points which are above the maximum potential operating temperature. The metallic oxides, commonly called refractory materials, are relatively inexpensive and readily available (with the exception of beryllium oxide). Beryllium oxide would be an excellant material to use were it not for its extremely high cost. Lithium oxide has the highest specific heat of any of the materials listed in Table 16. It is susceptible to chemical change, however, when exposed to atmospheric moisture at ambient temperature or to air at elevated temperatures.

In the second group of materials in Table 16 are some of the elements of low atomic weight and one compound. These latter materials also have sufficiently high heat capacities in the solid state to be attractive for thermal-energy storage systems. However, they are not as stable at temperatures above 2000 F as the refractory materials and therefore would have to be contained in such a way as not to come in contact with air or other reactive gases. They are also somewhat more expensive than the refractory materials.

Physical Characteristics – Thermal-Energy-Storage Systems

General System Configuration. Figure 25 is a simplified illustration of a basic thermal-energy storage system. The components of this system include: the heat-storage material and its container, referred to in combination as the thermal storage tank; the intermediate heat-transfer fluid loop with its circulating pump and the heat exchanger which supplies heat energy to the prime mover; and the recharging system with a combustion chamber, a fuel input line, and an exhaust system.

The container of the heat-storage material serves the important functions of insulating the material against excessive direct heat losses to the outside, preventing or minimizing deterioration of the material and any components in contact with it, restraining the material in whatever shape or form is desired for best performance, and providing the means for transferring heat to and from the material during operation and recharging. So the container is obviously more than just a box. In performing all these functions, the container and its associated insulation must be able to withstand high temperatures, must be lightweight, must be nonreactive, and should be low in cost.

TABLE 15. PERTINENT PHYSICAL CHARACTERISTICS AND CURRENT COSTS OF MATERIALS WITH HIGH HEATS OF FUSION AND QUALIFYING AS ATTRACTIVE THERMAL-ENERGY-STORAGE MATERIAL

Compound	Heat of Fusion, hp(t)-hr/lb	Specific Heat(a), hp(t)-hr/lb F	Melting Temperature, F	Density(b), lb/cu ft	Current Bulk Price, $/lb
Lithium Hydride (LiH)	0.432	0.000420(c)	1270	48.7	9.50
Lithium Fluoride (LiF)	0.176	0.000145(d)	1558	164.3	1.55
Lithium Hydroxide (LiOH)	0.149	0.000142(c)	884	88.0	0.54
Magnesium Silicide (Mg_2Si)	0.142	--	2016	--	--
Sodium Fluoride (NaF)	0.135	0.000110(e)	1854	168.4	0.14

(a) The listed values will be higher at higher temperatures.
(b) In solid state.
(c) At 122 F.
(d) At 50 F.
(e) At 212 F

TABLE 16. PERTINENT PHYSICAL CHARACTERISTICS AND CURRENT COSTS OF MATERIALS WITH HIGH THERMAL-ENERGY-STORAGE POTENTIAL IN SOLID STATE

Material	Specific Heat(a), hp(t)-hr/lb-F	Melting Temperature, F	Density lb/cu ft	Current Bulk Price, $/lb
Beryllium Oxide (BeO)	0.000176	4500/4550	188	12.00
Magnesium Oxide (MgO)	0.000133	5070	223	0.03
Aluminum Oxide (Al_2O_3)	0.000114	3650/3700	248	0.05
Silicon Dioxide (SiO_2)	0.000106	3110	145	0.0075
Titanium Monoxide (TiO)	0.000090	3180	307	0.25
Zirconium Dioxide (ZrO_2)	0.000059	4800/4900	352	0.48
Lithium Oxide (Li_2O)	0.000274	2600/2860	125	--
Boron Carbide (B_4C)	0.000184	--	158	--
Beryllium (Be)	0.000262	2341	115	--
Boron (B)	0.000204	4172	153	--
Carbon (C)	0.000145	>6400	140	--

(a) Average value for temperature range 500 to 2200 F.

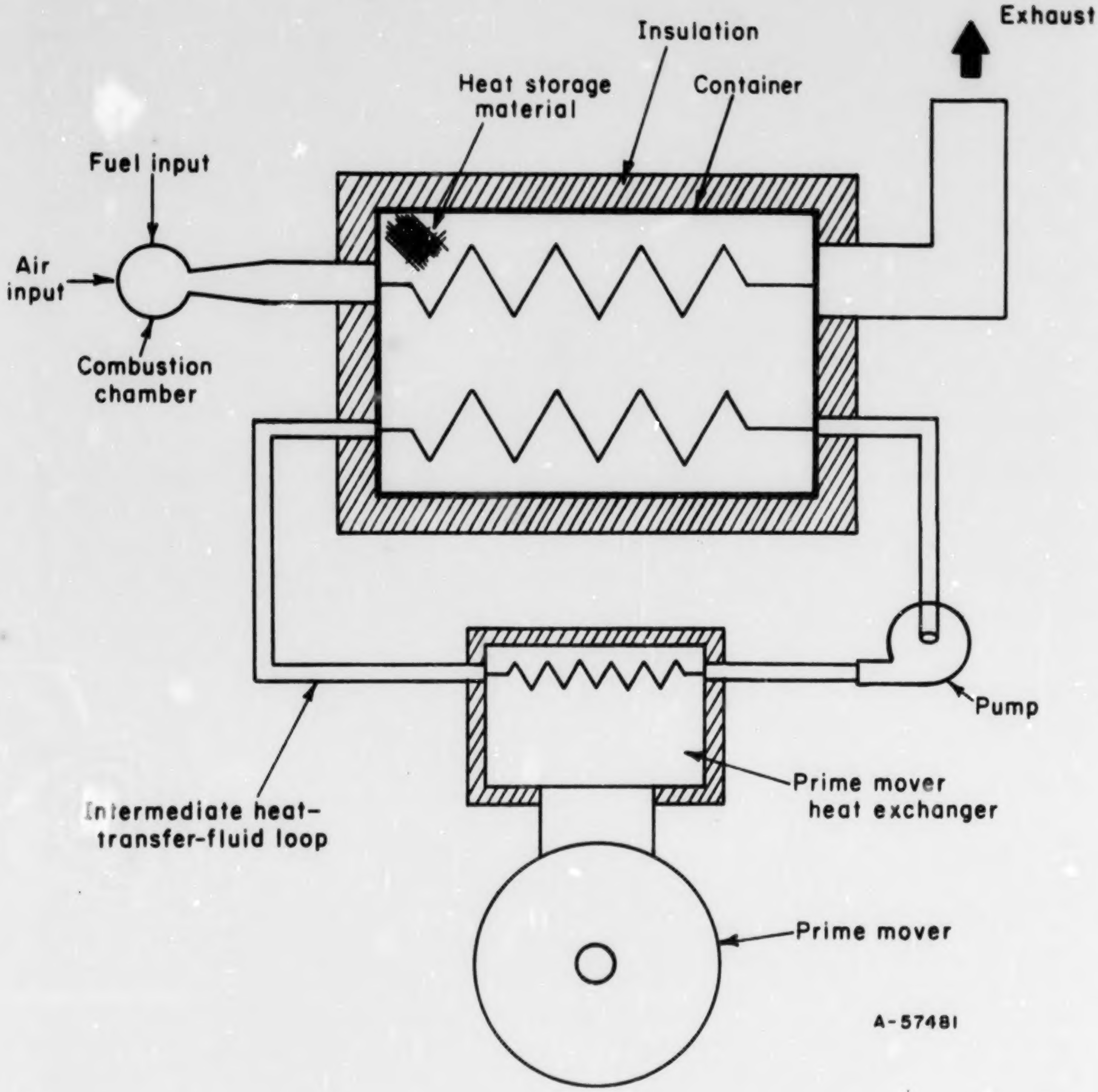

Exhaust
Insulation
Heat storage material
Container
Fuel input
Air input
Combustion chamber
Pump
Prime mover heat exchanger
Intermediate heat-transfer-fluid loop
Prime mover
A-57481

FIGURE 25. SCHEMATIC DIAGRAM OF BASIC THERMAL-ENERGY STORAGE SYSTEM

Two problems that must be solved when dealing with a molten heat-storage substance in a container are thermal expansion and corrosion. Most substances when changing from the liquid to the solid state usually either expand or contract. When a material expands in a confined space, the danger of rupturing the container wall or of crushing any heat-exchanger tubes imbedded in the material could be great. Lithium hydride and lithium fluoride contract upon solidyfying, while lithium hydroxide changes volume only slightly. These volume changes can generally be accommodated with careful design. A problem which can arise when the heat-storage material contracts is the problem of spalling or pulling away from the heat-transfer surface. The lithium fluroride and lithium hydroxide compounds have been shown not to flake off or pull away from the surface but appear to form a tight bond during the solidification process. Examples of materials which have been found to be reasonably compatible from a corrosion standpoint with the lithium compounds include nickel or a nickel-molybdenum alloy for lithium hydroxide, and Hastelloy N for lithium fluoride.

The design of a container for a refractory heat storage system could be considerably less critical than that for the molten heat storage system. Since there is no change of state involved in the contained material there would be no problem of expansion or contraction and chemical deterioration would be minimal or nonexistent. On the other hand, a refractory heat storage system might operate at a higher initial temperature; consequently the container might have to be lined on the inside with an insulating refractory material. Furthermore, the brittleness of refractory materials would impose a burden on the design and mounting of the system.

If a molten heat storage material is used in a thermal-energy storage system, the question might arise as to the possibility of pumping this material in the molten state directly to the prime-mover heat exchanger instead of using an intermediate heat-transfer fluid. Experience has shown, in sodium-cooled reactors for instance, that there are enough problems in trying to contain molten materials in one location at high temperature without trying to pump them through pipes to a remote location. Therefore, pumping the heat-storage material itself appears to be impractical for the passenger-vehicle application. A liquid metal for the intermediate heat-transfer fluid would be unacceptable for the same reasons. Water or any other conventional liquid could not be used because of the high temperatures and resultant high pressures. The best choice would appear to be a gas, and the most logical gas to use is air. If the circuit could be permanently sealed, then helium or even hydrogen would be preferable. However, a perfectly sealed system is considered impractical for this application.

A molten heat storage material having its entire mass contained in a single tank might conceivably maintain the same bulk temperature throughout as it cools, because of convection currents set up in the melt. Under these conditions, the outlet temperature of the intermediate heat-transfer fluid would not be higher than the average bulk temperature of the tank contents. On the other hand, if the system were designed such that the same total weight of heat-transfer material was dispersed in a number of individual containers with the intermediate heat-transfer fluid passing successively through these containers, then its outlet temperature could reach a value higher than the average temperature of the material in all of the containers since a temperature gradient would be set up from one container to the next. The second configuration, though probably considerably more complex, is the more desirable condition since the prime mover would be supplied with a higher temperature heat-transfer fluid over a longer period of time, thus permitting a greater overall energy-conversion efficiency.

The overall size and weight of the intermediate heat-transfer fluid loop, and the power required for pumping the air, can be significantly reduced by pressurizing the air. Pressures up to 100-psi guage might be appropriate in a passenger-vehicle system. The pressurization also provides a means for controlling the output of the thermal-energy storage system, for instance, reducing the pressure by bleeding air out of the system will reduce the heat output, and increasing the pressure will increase the heat output.

Recharging of the thermal-energy storage system can be accomplished in a number of different ways. One possible approach is to use products from the combustion of fuel, either passing the hot gases directly through the thermal storage tank, passing them over a heat exchanger mounted externally on the thermal storage tank, or using them to heat an intermediate heat-transfer fluid which is then pumped directly through the thermal storage tank. This combustion system could be integral with the thermal-energy storage system, as indicated in Figure 25, or it could be completely separate. Recharging could also be accomplished utilizing electrical energy such as in resistance heaters imbedded in the heat-storage material, or resistance heaters located in a heat exchanger external to the thermal-energy storage system and heating air which is then passed through the thermal storage tank.

System Design Studies. A number of design studies have been conducted for specific applications of thermal-energy storage systems. Table 17 is a compilation of available data from three such studies. The third study listed was recently completed by the General Motors Research Laboratories and represents an excellent projection of the physical characteristics that would be applicable to future vehicular thermal-energy storage systems.

TABLE 17. ESTIMATED PERFORMANCE OF PROPOSED AND EXPERIMENTAL THERMAL-ENERGY STORAGE SYSTEMS

Application	Reference	Material	System Size, hp(t)-hr	Specific Weight, lb/hp(t)-hr	Specific Volume, ft^3/hp(t)-hr	Specific Cost, $/hp(t)-hr
GM Research-Submarine	95	Aluminum oxide	66,700	5.3	0.045	1.0
GM Allison Div.-Satellite	96	Lithium hydride	11.6	3.9	0.240	2.0
GM Research-Vehicle	--	Lithium fluoride	100	3.3	0.032	--

GM Research - Submarine. This design study was undertaken on the basis that even though nuclear propulsion systems were proving to be extremely successful there would still be a need for submarines of small displacement, more limited submerged ranges, lower cost, and freedom from radiation hazards. The submarine size selected for the application of a thermal-energy storage system was slightly under 1500 tons submerged displacement with a horsepower requirement of 2,228 hp(s).

Total power-plant weight for this system was to be 484,000 pounds. The main components of the proposed power plant included: an aircraft-type gas turbine for surface or snorkel running and submerged emergency speed, two large Stirling engines for submerged cruising, one smaller Stirling engine for submerged creeping, and two even smaller Stirling engines for auxiliary electrical power generation, plus the necessary speed reducers, clutches, piping and insulation, expansion joints, and valves.

The function of the thermal-energy storage system was to provide the power for completely submerged operation. A recharging capability was included in the form of a liquid-fuel combustion system. The total weight of the Stirling engines, gas turbine, and other equipment mentioned above came to 129,000 pounds, leaving 355,000 pounds available for the thermal-energy storage-system weight. Aluminum oxide was selected as the heat-storage material to operate between an upper temperature of 2800 F and a lower temperature of 1350 F. The thermal storage tank consisted of a metal shell with a refractory insulation lining on the inside and a fibrous insulation on the outside. The tank, insulation, and combustion recharging equipment weighed approximately 46,000 pounds, and the heat-storage material weighed 309,000 pounds.

The thermal-energy storage system was capable of delivering a total stored energy of 66,700 hp(t)-hr. The heat loss from the thermal storage tank was estimated to be less than 0.05 percent per hour. Using this total available stored energy in the main propulsion Stirling engines, the submarine would be capable of sustaining a speed of 15 knots for slightly over 22 hours, which would give it a submerged range of about 340 nautical miles. At an emergency speed of 35 knots, and using the thermal-energy storage system stored energy in the gas turbine, the running time would be approximately 1-1/2 hours.

GM Allison Division – Satellite. This design study was conducted to determine the requirements for a thermal-energy storage system to be incorporated with a solar power system on board an earth satellite. This solar power system was designed to produce the heat energy required for a 4 hp(e) Stirling-engine-powered generator system. The basic power plant for this satellite consisted of a solar-energy collector and absorber, an intermediate heat-transfer fluid and pump, a heat reservoir, and a Stirling engine driving an electric generator. The heat reservoir was necessary because for the particular earth orbit mission envisioned for this satellite it would be in the earth's shadow for approximately 40 percent of each orbit, or about 35 minutes out of each 90 minutes.

The thermal-energy storage system proposed for this application consisted of conically shaped lithium hydride elements encased in electro-formed nickel or stainless steel jackets and disposed in a configuration allowing free passage of the intermediate heat-transfer fluid around each element. Minimum insulation was provided around the entire assembly because of the short charge and discharge cycle and because minimum weight was of paramount consideration.

A system sized for a 300-mile-high earth orbit was estimated to weigh a total of 45 pounds, including 15 pounds of the heat-storage material, lithium hydride, to occupy a space of about 2-3/4 cubic feet, and to be capable of producing about 11.6 hp(t)-hr from one charge. The authors concluded that a thermal-energy storage system based on these approximate specifications and performance data was technically feasible.

GM Research - Vehicles. General Motors is currently actively engaged in an investigation of the application of thermal-energy storage to passenger vehicles. Their current investigation was a natural extension of their studies for the submarine application, with the added impetus of the current search for a nonpolluting vehicle power plant. As a tangible demonstration of the technical feasibility of a thermal-energy storage system for vehicular application, a system was installed in a conventional Corvair automobile. This system consisted of a thermal-energy storage unit mounted in the front, trunk compartment of the vehicle and a Stirling engine located in the rear, engine compartment of the vehicle. Air was used as the intermediate heat-transfer fluid. The heat-storage material used was aluminum oxide in pellet form about 1 inch long, hexagonal in cross section, with a hole in the center of each and cutouts at the corners of the hexagon so that when properly stacked in the thermal storage tank there would be a multiplicity of air passages through the system. This was the same general configuration that was studied earlier for the submarine application.

The system worked and the vehicle performed reasonably well, amply demonstrating the technical feasibility of thermal-energy storage for vehicular use. No significant attempt was made to optimize the system either before or after installation in the vehicle; consequently, actual performance data from this application are not of direct interest to this study.

Further work in this area at General Motors, including both theoretical and laboratory studies, has resulted in an attractive second-generation concept for a vehicular thermal-energy storage power system. This concept is the third one listed in Table 17 and involves a unique and compact integration of the thermal-energy storage system and the prime mover. Using lithium fluoride as the heat-storage material and a Stirling engine as the prime mover, the key to this concept is having the Stirling-engine heater tubes buried right in the heat-storage material. Thus, no intermediate heat-transfer fluid with its associated hardware and losses is required.

The lithium fluoride is cycled within a temperature range from 1600 F maximum down to a cutoff temperature of about 750 F. In this temperature range both the heat of fusion and the sensible heat in the molten and solid states is utilized. With the engine heater tubes buried right in the lithium fluoride, a minimum temperature difference can be maintained between the heat-storage material and the working fluid of the Stirling engine.

Electric immersion heaters embedded in the lithium fluoride would supply the heat for recharging. A complete recharge from full discharge would take about 4 hours. Control of the engine power output and compensation for the decreasing lithium fluoride temperature would be accomplished by varying the working-fluid pressure within the engine. It is possible, by increasing the working-fluid pressure, to maintain a full-load maximum efficiency capability up to the point where about 85 percent of the stored thermal energy is consumed. During the last 15 percent of energy release, the engine efficiency and maximum power capability would taper off.

MECHANICAL ENERGY STORAGE

Three principal means of mechanical-energy storage have been considered: (1) the storage of kinetic energy in flywheels, (2) the eleastic deformation of solids (springs), and (3) the compression of a gas, namely in terms of a hydraulic accumulator system. A discussion of each of these follows.

Flywheels

In the past few years there has been a renewal of interest in flywheels as sources of stored power. Their use is most appropriate in applications where many charge and discharge cycles are contemplated, and where the energy does not have to be stored for a long period of time. They have the desirable feature that all of their stored energy can be removed rapidly (i. e. , they can be designed to have a high specific power). These characteristics make them natural candidates for hybrid vehicle propulsion systems, where they would be used in conjunction with an engine of reduced size, and more importantly for this study, of reduced pollutant emissions.

General Description

A flywheel for automotive application would most likely be a steel disk thicker at the hub than at the rim. It would be enclosed in a housing that would be evacuated to a pressure of 0. 1 to 0. 01 atm by a small vacuum pump. Pressure-lubricated ball bearings would probably be required. The wheel and its auxiliaries present no serious technical problems.

The difficult problem is how to couple the flywheel to the vehicle wheels in an effective and inexpensive way. A steplessly variable drive is desired, since the flywheel must decelerate smoothly as the vehicle accelerates. The candidate systems include hydrostatic and electrical drives, traction drives, special torque converters, and planetary gearsets which allow changes in flywheel speed to be compensated by changes in engine speed. The torque converter is perhaps the lowest cost system, but it requires development to meet the unusual operating conditions. The hydrostatic system is perhaps the most desirable in regard to performance and size. The planetary gear system is an interesting compromise whose possibilities have not been completely explored.

Current and Projected State of the Art

Existing Units and Applications. Several applications in which flywheels have been used for moderately long-time energy storage are listed in Table 18. Of these, the Oerlikon Electrogyro is the only application to vehicle propulsion that was put into commercial service. It was a transit bus that had no engine. When the bus stopped to take on passengers, electrical contact was made with an overhead terminal and energy was imparted to the flywheel. These buses were used in several European and African cities, but are no longer being built. Even though the flywheel used was a very large, low-performance one, the failure of the Electrogyro is attributed to operating inconveniences resulting from lack of an engine rather than to any technical difficulty with thee flywheel.

TABLE 18. EXISTING UNITS AND APPLICATIONS OF FLYWHEELS FOR ENERGY STORAGE

System	Reference	Application	Wheel Type and Weight	Energy Stored, hp(s)-hr	Specific Weight[(a)], lb/hp(s)-hr	Drive System	Status
Oerlikon Electrogyro	97	Transit bus	Steel rim, 3300 lb	12	275	Electric	No longer being built
Gyreacta and Hydreacta	--	Vehicle transmission	Steel disk, 230 lb	--	--	Planetary	Development suspended?
General Dynamics	98	Power supply for aerospace use	Glass fiber, 56 lb	1.5	38	Electric	Apparent success
[illegible] Aviation	99	[illegible] storage for aircraft control functions	Steel disk, widely variable	1.7 for 100-lb wheel	59	Hydrostatic	Under development

(a) Wheel only.

The Gyreacta transmission is a more modern, higher performance device that augments the performance of an engine. The flywheel and planetary gearbox are incorporated into an attractively compact unit that is similar in appearance to a passenger car automatic transmission. The gearbox is quite complicated, however, and to the best of our knowledge the developer has not placed this unit on the market.

Physical, Performance, and Cost Characteristics. The effectiveness of a flywhee in storing a given quantity of energy in a unit of minimum size and weight depends upon the material of the wheel and its general configuration, i. e., whether rim, disk, or specially contoured. The conventional design of a rim with spokes or a relatively light webb is best only if there are simultaneous limitations on the diameter and the rotative speed such that the centrifugal stress in the material does not limit the design. When there is a choice of speed, a disk can be run approximately 50 percent faster than a thin rim of the same diameter, because it retains its material primarily by radial tension rather than by hoop tension. Although heavier, the disk will store approximately 20 percent more energy per pound of flywheel material. As the rim is made thicker in the radial direction, the advantage of the disk increases. The limiting case of a thick rim is a disk with a very small hole in the center. This will store just half as much energy per pound as the unpierced disk.

Because of the centrifugal stresses in a disk are greatest at its center, they can be significantly reduced by the addition of a relatively small amount of material in the form of added axial thickness at this point. The wheel can then be run faster, and will store still more energy per pound. The most effective wheel on an energy-per-pound basis is thought to be one shaped so as to approximate a condition of uniform centrifugal stress throughout the disk. Some wheels of this type have been made with rim thicknesses only 5 to 10 percent of the thickness near the hub. As a practical device, however, the constant-stress wheel is a victim of diminishing returns. The housing required to enclose its large diameter, thin rim weighs far more than that portion of the wheel, and the large area at high speed increases the aerodynamic drag loss. The most practical configuration for use in a vehicle is thought to be a disk with the hub thickened by 50 to 100 percent to reduce the stress.

Material. If energy stored per pound of flywheel is taken to be the criterion of excellence, and if simultaneous limitations on speed and diameter do not prevent working the material to its allowable stress, then the effectiveness of a flywheel material can be shown to be directly proportional to its strength-to-weight ratio. When choosing among the metallics, therefore, there is little motivation for use of anything but steel forgings. Very high performance steel wheels for aircraft applications are presently being designed for working stresses approaching 140,000 psi. Stresses on this order can be contemplated because of the accuracy with which the loading is known. For vehicle applications, however, it is anticipated that the safety measures required would make wheels working much above 100,000 psi uneconomical.

In the area of the nonmetallics, there are a number of modern materials with strength-to-weight ratios considerably better than those of steel. The most attractive of these at present is a composite of glass fibers wound in an epoxy filler. Although its strength-to-weight ratio is about three times that of steel, this material is not really that much better because it is nonhomogeneous and must, at the present state of the art, carry its stresses as a hoop. The weight saved over that of a constant-stress steel wheel virtually disappears unless the glass-fiber rim is made quite thin. Since the glass-fiber wheels are also expensive, bulky, and limited in their ability to transmit high torque to the hub structure, they are not recommended except for very high performance aerospace applications, where the ambient vacuum makes it unnecessary to house them. More advanced materials utilizing fibers of materials such as boron, graphite, and silicon carbide(100) do offer a possible performance advantage over steel. For the foreseeable future, however, the associated basic material and wheel-fabrication costs will be far too high relative to potential performance advantage for automotive applications.

Aerodynamic Drag. Energy lost to windage becomes an increasingly important component of inefficiency as design practice tends toward smaller, higher speed wheels. An 18-inch disk operating at 20,000 rpm in atmospheric air will dissipate about 20 hp(s). By evacuating the housing to 0.01 atm, however, the loss can be reduced to 0.5 hp(s).

Gyroscopic Effects. The trend toward smaller, higher speed wheels has resulted in a reduction in the gyroscopic torques generated. It is not anticipated that gyroscopic torques will be large enough to significantly affect the handling of any ordinary road vehicle. It is necessary, however, to design the flywheel bearings with due regard for forces generated in response to sudden motions imposed by the vehicle suspension system on a rough road.

Specific Weight, Volume, and Cost. Attainable specific-weight figures for a few appropriate combinations of material and shape are given in Table 19. The weight and occupied volume of the housing required have also been estimated, but these should be regarded as very rough preliminary figures. The bearing assembly and vacuum pump are included, but not the gear reduction. For automotive application, a specific weight and a specific volume of approximately 100 lb/hp(s)-hr and 0.25 ft^3/hp(s)-hr, respectively, appear to be most appropriate. These values would apply to a wheel designed for a stress level somewhat below 140,000 psi and having a basic disk configuration with the hub thickened by 50 to 100 percent. Assuming a minimum O. E. M. or dealer's cost of approximately $1/lb, the estimated specific cost would be $100/hp(s)-hr.

TABLE 19. SPECIFIC WEIGHT AND VOLUME OF FLYWHEELS AND HOUSING

	Specific Weight, Wheel Only, lb/hp(s)-hr	Estimated Weight With Housing, lb/hp(s)-hr	Estimated Occupied Volume of Housing, cu ft/hp(s)-hr
Constant-thickness steel disk, 100,000 psi	111	132	0.32
Constant-thickness steel disk, 140,000 psi	79	94	0.23
Constant-stress steel wheel, 140,000 psi	58	78	0.43
Glass-fiber rim, 90 percent yield stress	38	76	1.27

Springs

Although springs are the most commonly used energy-storing devices, they are presently used only where relatively small amounts of energy are to be stored. The storage of quantities of energy large enough to accelerate a vehicle requires an advance in the state of the art in that springs having superior specific-weight and specific-volume characteristics must be developed. This means that steel would be replaced by elastomerics as the spring material. The development would be a difficult one, particularly in regard to life of the spring elements and efficiency of energy recovery.

General Description

The energy storage unit for use in a vehicle would no doubt consist of a multiplicity of small springs rather than one large one. This would make it easier to manufacture, would make possible fuller development of potential material properties, and would allow attractive packaging configurations. In addition to the springs themselves, a mechanism would be required for combining the forces and motions of the many springs into a single output, a speed-increasing gear train, and a variable-ratio drive to match the varying torque requirements of the vehicle to the varying torque delivery of the springs. The combining and speed-increasing requirements would be simplified if the springs were mounted in series, so that their deformations would add.

Current and Projected State of the Art

Existing Applications. No pertinent applications of springs are known, nor are there applications of elastomerics with the intent of storing appreciable quantities of energy for reuse. Energy storage springs presently in use are of steel and are mostly of the spiral-motor or the Belleville type. If made large enough to provide power for a road vehicle, they would be far too heavy.

Physical, Performance, and Cost Characteristics. The most important indicator of performance for a vehicle propulsion spring has been identified as the ratio of the energy stored to the weight of the spring material storing it. Most common springs are so heavy that they do not store enough energy to accelerate their own masses to highway speed. For materials with linear stress-strain curves, the energy stored per pound is proportional to

$$\frac{S^2}{M\rho} ,$$

where

S = working stress

M = appropriate modulus of elasticity

ρ = density.

Either shear or tensile effectiveness may be calculated by using the appropriate modulus and working stress. Elastomers usually have seriously nonlinear stress-strain curves, making it necessary to estimate the area under the curve to determine the energy stored. Effective use of any material depends, of course, on stressing all of the material used to the same extent.

Metallic Springs. Among the metallics, the above index of merit indicates steel to be the best spring material. Steel in tension or compression is about 20 percent better than steel in shear, but is not recommended because of the difficulty in handling the small deflections and large forces that result from pure tensile loading. Uniform shear can be conveniently applied by using the material in the form of a thin-walled tubular torsion bar. Such a tube in torsion can store twice as much energy per pound as a solid round in torsion (a conventional helical compression or tension spring), and can store about five times as much energy per pound as a rectangular bar in bending (a conventional leaf or spiral motor spring). It is also better than a Belleville washer, which has similar difficulties in working all its material to capacity. Spiral motor springs and

Belleville washers have been favored in the past because the force delivered remains relatively constant over a large deflection range. Where really high energy storage is required in a light package, this is a luxury that cannot be afforded. Changes in force must be compensated for by the driveline.

Elastomeric Springs. Among the elastomerics, natural rubber and urethane have the highest energy-storage potentials per pound of material. Neoprene rubber is the only other material that comes close to these two. Energy storage is virtually the same in tension and in shear, since the deformation amounts to several hundred percent, causing the material to be basically in tension in either case.

Rubber springs are loaded by bonding opposite faces to rigid metal parts which move relative to one another. It is very important to avoid high tensile stresses at the edge of the bond area, since, otherwise, peel failure will prevent full loading of the rubber. For this reason, tubular sections in torsion are recommended over tensile arrangements or shear pads. The shear rate may be applied radially by an internal bar and an external sleeve that rotate relative to one another, or it may be applied axially by two end plates with relative rotation. The latter arrangement seems particularly well suited for large-capacity springs, since many sandwiches could be stacked end to end.

Urethanes cannot be loaded in the same way that rubbers are, since strong adhesive bonds cannot be made to them. A mechanically applied tensile load would probably be used. Considerable development would be required here.

Specific Weight and Volume and Bulk-Material Costs. Comparative performance and cost figures for springs of steel, natural rubber, and urethane in appropriate loading conditions are given in Table 20. These figures cover only the spring itself, and not the driveline components. The estimates are nonconservative, the steel being worked to its endurance limit and the elastomers to their breaking points.

It can be seen that the steel spring is more than 100 times heavier than the elastomeric ones for a given quantity of energy stored. It is so heavy, in fact, that the energy stored will accelerate the mass of the spring to only 56 mph. Consequently, any vehicle that is driven solely by a steel spring which moves with the vehicle is limited by theory to a top speed of less than 56 mph. A vehicle with an engine in addition to the spring could, of course, go faster, but it appears evident that the amount of energy storable would not be sufficient to appreciably affect performance or economy. Steel is not, therefore, a candidate material for a vehicle propulsion spring.

If the elastomers are now compared, it can be seen that the urethane springs are somewhat smaller and lighter than those of natural rubber. It is believed, however, that this advantage will be more than offset by the difficulty in loading the urethanes and by their higher cost. In addition, the urethanes have a potential hysteresis problem that results in inefficient recovery of the energy stored. The amount of hysteresis can be changed greatly by adjustment of the formulation of the urethane. Attempts to increase the hysteresis for applications where damping is desired has been very successful. It is not known how much effort has been expended on attempts to decrease the hysteresis.

TABLE 20. PROPERTIES OF SPRING MATERIALS

	Weight of Spring Material Only[a], lb/hp(s)-hr	Volume of Spring Material Only[b], cu ft/hr(s)-hr	Approximate Efficiency, percent	Current Material cost[c], $/lb
Steel in shear, 90,000 psi	19,100	39.0	99	
Natural rubber, unfilled, in shear	93	1.36	90	$0.20
DuPont LD-167 urethane, Shore A 95, in tension	69	1.02	75	1.00
Acushnet Elastacast polyurethane, Shore A 85, in tension	58	0.82	--	1.00

(a) Weight of system would probably be 50 to 100 percent greater than this.
(b) Volume of system would probably be 50 to 100 percent greater than this.
(c) Approximate cost of basic material purchased in large quantities. Cost of part fabrication not included.

On the basis of the above discussion it is concluded that the most likely spring design would be natural rubber fabricated in tubular sections and loaded in torsion.

Compression of Gases

General Description

The most likely system for utilizing the compression of a gas as a means of storing energy for automotive application would be one utilizing a hydraulic accumulator. Such a system might consist of a spherical accumulator with a flexible diaphragm to separate the gas from the hydraulic fluid. The gas and hydraulic fluid would both be at the same high pressure. When the compressed gas expanded, or released energy, the hydraulic fluid would be forced out of the accumulator and through a variable-displacement, hydrostatic "pump-motor". As the hydraulic fluid passed through the motor it would experience a drop in pressure and would drive the pump shaft, which would be geared into the drive train of the vehicle in question. After passing through the motor, the hydraulic fluid would be collected in a low-pressure reservoir.

To store energy in the gas, the hydraulic "pump-motor" would be driven as a pump and would pump the low-pressure fluid in the reservoir back into the high-pressure accumulator. This would result in a compression of the gas and storage of energy in the accumulator.

Current and Projected State of the Art

To investigate the characteristics of such a system, a number of optimistic assumptions were made. First it was assumed that the accumulator would be a uniform-wall-thickness sphere made of steel with a permissable design stress of 100,000 psi. Maximum accumulator pressure was assumed to be 5000 psi. While this pressure is higher than that used in the great majority of industrial hydraulic systems, some aircraft applications and a few industrial applications operate at pressures this high or higher.

With these assumptions as a starting point, an analysis then was conducted to determine the optimum gas-expansion ratio from the stand points of both weight and size. That is, what gas-expansion ratio would result in the maximum number of hp(s)-hr being stored per pound of accumulator, including the weight of both the steel sphere and hydraulic fluid? Likewise, what expansion ratio would result in the largest number of hp(s)-hr being stored per cubic foot of accumulator?

The results of this analysis are listed in Table 21. The optimum expansion ratios were determined, assuming both an isothermal and isentropic expansion process. When optimizing for weight, the optimum expansion ratio calculated on an isentropic basis was 14 percent less than that calculated on an isothermal basis. Likewise, when optimizing for volume, the ratio calculated on an isentropic basis was 21 percent less than that calculated on an isothermal basis. Since the amount of energy available during an isothermal expansion is greater than that for isentropic expansion, the former was assumed to exist, and the specific weight and volumes for the accumulator and fluid were calculated on this basis.

TABLE 21. SPECIFIC WEIGHT AND VOLUME OF HYDRAULIC ACCUMULATORS FOR COMPRESSED-GAS-ENERGY STORAGE(a)

	Expansion Process	Optimum Expansion Ratio	Specific Weight(b), lb/hp(s)-hr	Specific Volume(c), ft^3/hp(s)-hr
Expansion Ratio Optimized for Weight	Isothermal	2.06	495	7.86
	Isentropic	1.77	--	--
Expansion Ratio Optimized for Volume	Isothermal	2.72	515	7.50
	Isentropic	2.15	--	--

(a) Hydraulic accumulator assumed to be uniform-wall-thickness sphere made of steel with a permissable design stress of 100,000 psi. Maximum internal pressure assumed to be 5000 psi.

(b) Weight of steel sphere and hydraulic fluid only.

(c) Volume of accumulator only.

VEHICLE TRANSMISSIONS

(MECHANICAL, HYDROKINETIC, HYDROSTATIC)

In this discussion, the transmission is defined as a variable, mechanical-shaft-power converter for the conversion of mechanical-shaft power at one condition of speed and torque to a variety of conditions of speed and torque. Its primary functions are to provide a means of (1) uncoupling the engine for starting and idling purposes, (2) varying the torque and speed ratio betwen the engine and driven wheels as required, and (3) changing the direction of vehicle motion. For certain transmissions and applications it is the additional function of the transmissions to provide a vehicle braking capability by converting mechanical-shaft energy into thermal or some other form of energy.

General Description

Vehicle transmissions are usually divided into four principal categories: mechanical, hydrokinetic, hydrostatic, and electric. Each of these principal categories in turn is composed of a myriad of specific transmission concepts. For example, under mechanical transmissions would be classified positive engagement, friction drive, and viscous drive devices. These classifications can in turn be broken down into numerous specific concepts. For the purposes of this study, however, only the most advanced concept with respect to automotive application is discussed for each category. Also, only the first three categories fall within the scope of this program and are discussed in the following paragraphs.

Mechanical Transmissions

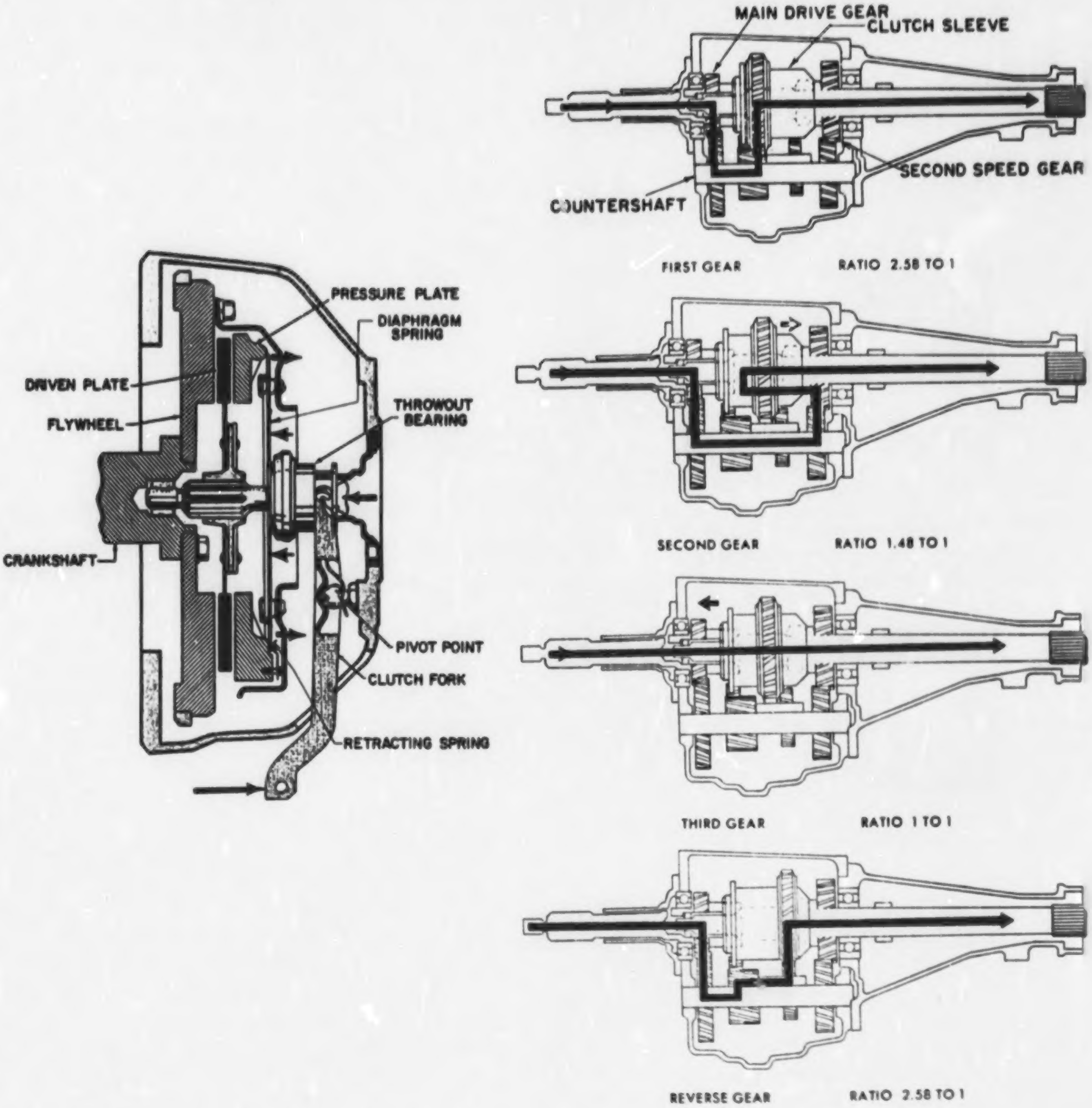
Figure 26 shows the arrangement of components in a typical 3-speed mechanical transmission clutch and gearbox. Power is transmitted from the engine flywheel to the main drive gear by means of the dry, single-plate friction disk, and in this particular unit, normal force for the friction elements is supplied by a diaphragm-type spring. During gear changes, the clutch fork is actuated in the direction indicated; this causes the diaphragm-type clutch spring to bow and permits the retracting springs to draw the friction elements out of contact. With the clutch disengaged, the desired gear ratio is selected by means of a shifting linkage which slides one or two gears along a splined shaft to produce a particular combination of mesh. With the exception of the direct-drive condition, power flow always passes through the main drive gear into the countershaft, and then back to the clutch sleeve through one of the gear combinations and a splined connection. Small friction clutches known as synchronizers are commonly interposed between the clutch sleeve and the main drive and second-speed gears to match the speed of these elements just before they are brought into mesh.

Variations of this basic arrangement include the use of helical clutch springs in place of the diaphragm type-spring shown, the addition of more gear ratios, and the incorporation of overdrive units. In heavy-duty trucks, some two-plate clutch designs are used, and 10, 12, and 15-speed transmissions are not uncommon. In the case of a 10-speed transmission, the 10 ratios are usually obtained by integrally mounting a 2-speed reduction unit in series with a 5-speed unit.

Hydrokinetic Transmissions

Hydrokinetic transmissions, often referred to as automatics, are installed in approximately 80 percent of currently built American passenger cars. As shown in Figure 27, these transmissions consist of two basic sections - a torque converter and a planetary gear box. The converter section is filled with hydraulic fluid, and power is transmitted from the engine flywheel to the input shaft of the planetary gear box by means of the hydraulic coupling action of the converter. Engine torque applied to the converter pump forces an energy transmitting flow of oil into the converter turbine - the turbine, in turn, absorbs this kinetic energy and converts it into a torque acting on the planetary input shaft. Manual and/or automatic actuation of the transmission controls then locks or engages various elements of the planetary gear set to provide two definite forward gear ratios, one reverse ratio, and one neutral position.

Common variations of the arrangement shown in Figure 27 include the use of two planetary gear sets to obtain 3 or 4 forward-speed ratios; the use of converter lock-out clutches to provide a positive mechanical input for certain gear ranges and/or cruising operation; and the use of multiple and/or variable converter elements to provide certain desirable operating characteristics for particular applications. Various combinations of band and multidisk clutch units are applied for engaging the planetary gear elements, and some differences also exist in the hydraulic circuits which control the shifting sequence. The latter usually involve mechanical or hydraulic governor devices which sense engine and vehicle speed, and another element which utlizes engine manifold vacuum and throttle position as a measure of operating load. It should be noted that the torque converter is also a torque multiplier, and that individual converter designs are tailored to provide the desired combination of torque multiplication and responsiveness.


MAIN DRIVE GEAR
CLUTCH SLEEVE
SECOND SPEED GEAR
COUNTERSHAFT
PRESSURE PLATE
DIAPHRAGM SPRING
DRIVEN PLATE
FLYWHEEL
THROWOUT BEARING
CRANKSHAFT
PIVOT POINT
CLUTCH FORK
RETRACTING SPRING

FIRST GEAR RATIO 2.58 TO 1

SECOND GEAR RATIO 1.48 TO 1

THIRD GEAR RATIO 1 TO 1

REVERSE GEAR RATIO 2.58 TO 1

FIGURE 26. MECHANICAL TRANSMISSION CLUTCH AND GEAR BOX

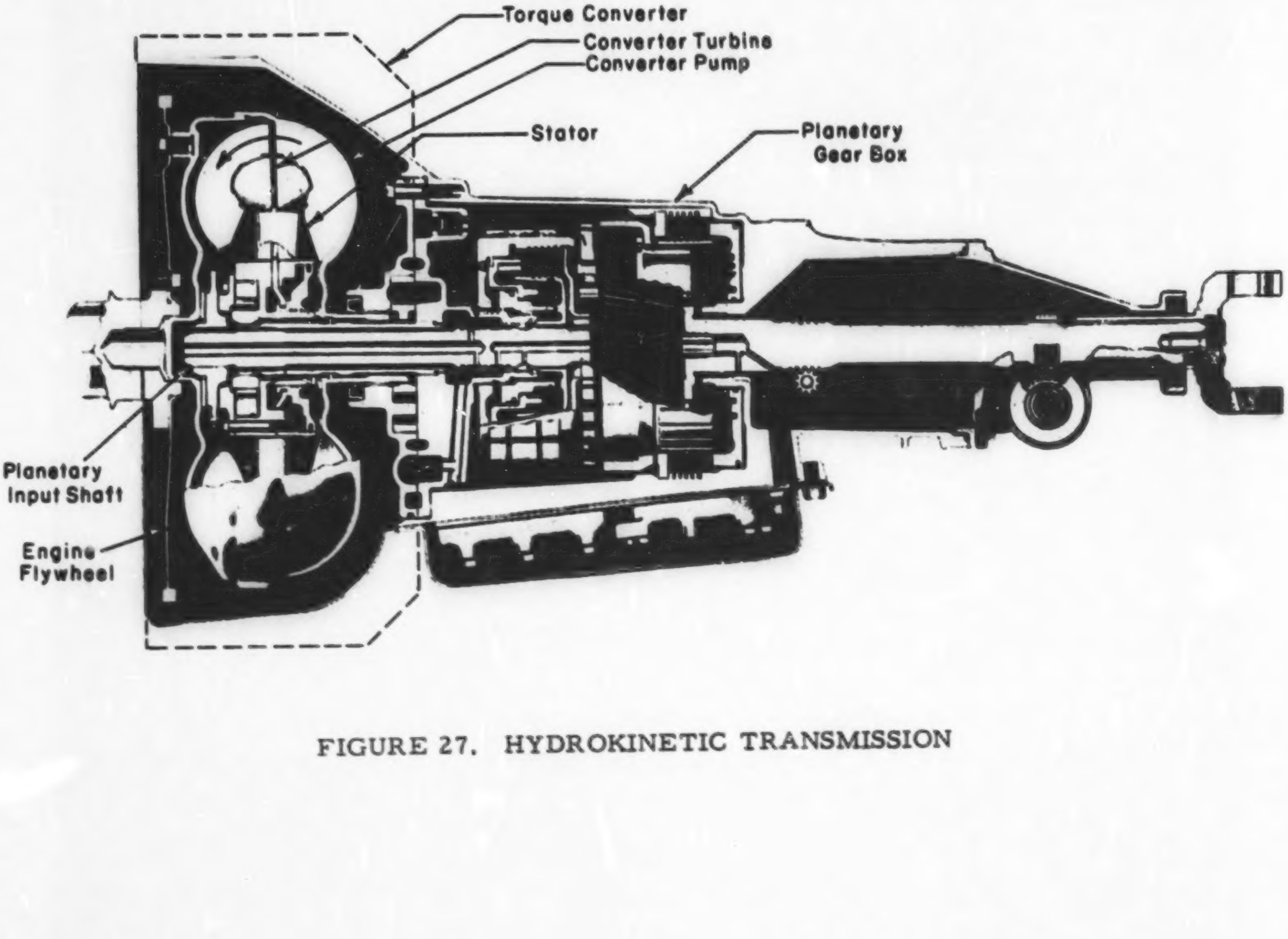

Torque Converter
Converter Turbine
Converter Pump
Stator
Planetary Gear Box
Planetary Input Shaft
Engine Flywheel

FIGURE 27. HYDROKINETIC TRANSMISSION

Hydrostatic Transmissions

A hydrostatic transmission consists essentially of one or more positive-displacement hydraulic pumps connected by pressure lines to one or more positive-displacement hydraulic motors. Displacement of the pump and motor may be either fixed or variable, and the pump and motor may or may not be of identical construction. By definition, hydrostatic transmissions differ from hydrokinetic transmissions in that power is transmitted by means of hydraulic pressure as opposed to fluid velocity.

Hydrostatic transmissions have no typical configuration, but their principle of operation may be visualized by the inspection of Figure 28. In this particular arrangement, a variable-displacement axial-piston pump is used to drive two fixed-displacement axial-piston motors. A neutral or no-drive condition is obtained by setting the variable-displacement pump for zero flow. Gradually tilting the movable swash plate in one direction then causes a gradual increase in forward vehicle motion, and tilting in the opposite direction produces a gradual increase in reverse motion.

Variations of this basic arrangement are limitless. All of the positive-displacement pump types (axial-piston, radial-piston, vane, and gear) may be used in various combinations; the motor and pump units may be "end-to-end" (as shown in Figure 28) or remotely located and conncected by tubing or hoses; and combinations of variable displacement units, motor controls, and/or integral gear elements can be arranged to provide a wide variation in torque and speed chracteristics to match particular applications.

Current and Projected State of the Art

Properties for a sampling of current vehicle transmissions are shown in Table 22. As indicated by the column headings, many of these values are estimates. Data sources included product literature; manufacturers and/or their representatives; local users; and technical reports, texts, and articles. Data which were not available are indicated by the notation (n. a.). General conclusions which may be drawn from these data are as follows:

(1) For a given class of vehicles, a mechanical transmission offers the highest peak efficiency, the lowest unit cost, and the smallest envelope volume.

(2) For passenger cars, incorporation of a hydrokinetic transmission in place of a mechanical transmission results in a relatively small weight and efficiency penalty and a relatively large cost penalty.

(3) Hydrostatic transmissions are used principally in off-highway equipment where their lower efficiency and high initial cost are offset by their unique installation and operating characteristics.

(4) Cost, weight, and size figures for industrial units are 5 to 10 times higher than those for corresponding passenger-car units. This difference occurs as a function of the number of units produced, the higher number of speed ratios or ratio spread provided, and the construction requirements for different duty cycles.

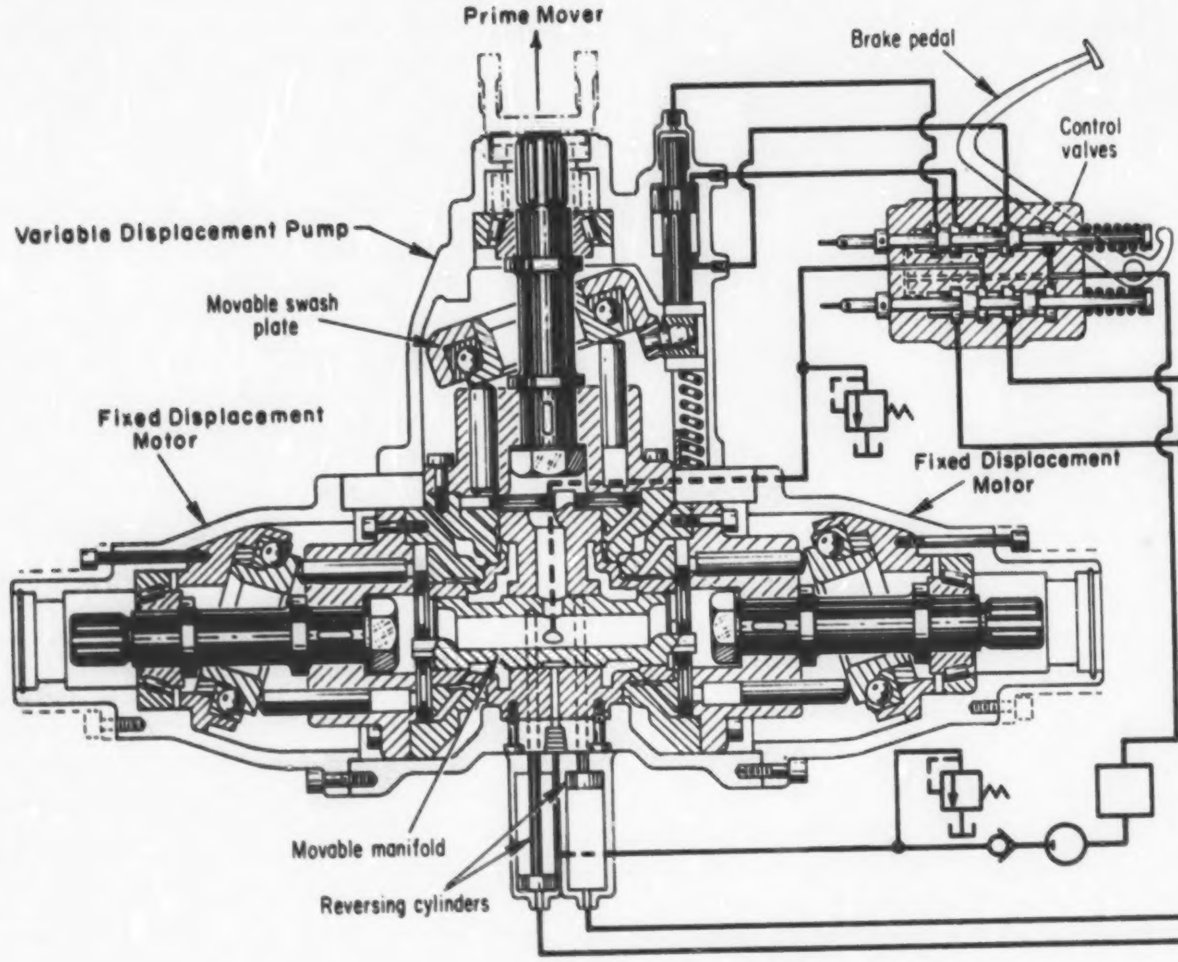

Prime Mover
Brake pedal
Control valves
Variable Displacement Pump
Movable swash plate
Fixed Displacement Motor
Fixed Displacement Motor
Movable manifold
Reversing cylinders

FIGURE 28. HYDROSTATIC TRANSMISSION

All three of these mechanisms are highly developed, with hydrostatics being the least so in this respect. While improvements will continue to be made in these transmissions, they will not be of a nature to permit estimates of "1980" specific parameters to deviate significantly from those listed in Table 22. This is particularly so when considering the fact that the transmissions of interest to this program are generally of lower power than those listed in Table 22. The principal exception to the preceding is the case of high power, high pressure, ultrahigh speed, hydrostatic systems.

NUCLEAR DEVICES

With respect to nuclear devices, the two major sources of nuclear energy, fission reactors and radioisotopes, and two direct nuclear-to-electric energy-conversion devices, alpha- and betavoltaic cells, were studied to determine whether they would be applicable for use in urban-vehicle propulsion systems in the next 10 to 15 years. The alpha- and betavoltaic cells would most likely be used with a radioisotope energy source; therefore, in the following paragraphs, radioisotopes are discussed in connection with both thermal and direct electrical power generators.

One vehicle-propulsion-system consideration that is unique to nuclear devices is the fact that to have radioactive material in a highly populated area would require adequate shielding and many safety precautions. The allowed radiation dose for the general public is 0.5 roentgen equivalent man (rem) per year or approximately 1.4 mrem/day. To protect passengers, pedestrians, and the general environment from a harmful radiation dose, the shielding would have to be "4 π", that is, completely surrounding the radioactive source. Also, the shield would have to withstand any conceivable major accident.

Reactor Systems

Three reactor systems typical of portable or semiportable systems were investigated. These are the ML-1 military reactor and the SNAP-8 and SNAP-50/SPUR space reactor systems. The ML-1 is gas-cooled and the SNAP-8 and the SNAP-50/SPUR are liquid-metal cooled. Water-cooled reactor systems were not investigated because of their inherently large size.

ML-1 System

General Description. The ML-1 is a mobile power plant developed for the United States Atomic Energy Commission and the United States Army Corps of Engineers by Aerojet-General Corporation. It consists of a nuclear reactor which supplies heat energy to a closed Brayton-cycle (gas turbine) engine, which in turn drives an electric alternator. Oxygenated nitrogen is used as the working fluid in the closed Brayton-cycle engine. The power plant is rated to produce a net output of 400 hp(e) at an ambient air temperature of 100 F and 670 hp(e) at -65 F.

TABLE 22. PROPERTIES OF

Manufacturer and Model	Principal Applications	Estimated Actual Power Capacity, hp(s)	Input Speed at Max Power, rpm	Input Torque at Max Power, ft. lb
				Mechanical
General Motors Corporation 3 Speed Manual	Full Size Passenger Car	270	4500	315
Ford Motor Company 3 Speed Manual	Compact Passenger Car	200	4300	244
Fuller Transmission Div., R-96, 10 Speed Manual	Heavy Duty Linehaul Truck	284	2100	710
Fuller Transmission Div., RT-910, 10 Speed Manual	Heavy Duty Linehaul Truck	340	2100	850
				Hydrokinetic
General Motors Corporation, 3 Speed Turbo-Hydra-Matic	Full Size Passenger Car & Light Duty Delivery Vans	300	4700	335
Ford Motor Company, 3 Speed, C-4 Cruise-O-Matic	Compact Passenger Cars and Econoline Vans	200	4300	244
Allison Div., GMC 6-Speed Transmatic	Cross Country Bus Heavy Duty Trucks	300	2100	750
Allison Div., GMC 6-Speed Transmatic	Linehaul Trucks	180	2000	472
				Hydrostatic
Dynex (Hydro-Planetary Hub)	Heavy Duty-off Highway Vehicles	400	2500	840
Eaton, Yale & Towne (Tacrodyne 2/60)	Tractors, Fork-Lift Trucks, Dump Trucks, etc.	60	2500	126
Sta-Rite Industries (C-3 Series Hydrostatic)	Machine tools, farm accessories	30	3000	52.5
Sundstrand	Tractors, Etc. Small Garden Tractors	81.4 12	~3000 ~3200	142 19.7

CURRENT VEHICLE TRANSMISSIONS

Ratio Spread	Weight, lb	Estimated Specific Weight, lb/hp(s)	Envelope Volume, ft^3	Estimated Specific Volume, $ft^3/hp(s)$	Estimated Dealers Cost, $	Estimated Specific Cost, $/hp(s)	Estimated Peak Efficiency, percent
Transmissions							
2.41:1	112.5	0.4	0.90	0.003	69	0.3	95
2.99:1	99	0.5	0.76	0.004	64	0.4	
9.65:1	820	2.9	7.07	0.025	1,800	6.3	
8.05:1	706	2.1	7.15	0.021	1,950	5.7	
Transmissions							
5.06:1	185	0.6	1.30	0.004	210	0.7	90
4.97:1	145	0.7	0.80	0.004	195	1.0	
18.5:1	990	3.3	10.15	0.034	3,600	12.0	
15.8:1	530	2.9	6.31	0.035	2,500	13.9	
Transmissions							
18:1	n.a.	--	n.a.	--	8,000	20.0	80
10:1	194	3.2	2.44	0.04	n.a.	--	
n.a.	50	1.7	0.52	0.017	265	8.8	
n.a. n.a.	n.a. n.a.	-- --	n.a. n.a.	-- --	676 120	8.3 13.0	

The reactor is a heterogeneous, water-moderated system, fueled with enriched uranium dioxide. An integral lead radiation shield surrounds the reactor to protect personnel during reactor shutdown. The core, shielding, and pressure-vessel assembly are enclosed in a tank of borated water during operation to provide additional neutron shielding.

Major components of the power-conversion system are a turbine-compressor set, reduction gear, alternator and starting motor, precooler with fans, recuperator, switch gear, and connecting piping and valves. The hot gas leaves the reactor at 1,200 F; expands in the turbine; passes through the low pressure side of a regenerative heat exchanger (recuperator) and through an air-cooled precooler where the waste heat is rejected to the atmosphere. After being compressed in the compressor, the gas is preheated to about 800 F as it passes through the high-pressure side of the recuperator. It then flows through the reactor to the turbine inlet, completing the cycle. The turbine is direct-coupled to the compressor and drives the alternator through a gear box.

Physical, Performance, and Cost Characteristics. The total weight of the ML-1 system is approximately 40 tons(101). The reactor alone weighs 30,000 lb. The power-conversion equipment weights 30,000 lb, the control cab 5,000 lb, and the auxiliary equipment 12,000 lb. The specific-weight requirement for the entire system was set at 150 lb/hp(e).

Because of radiation problems, however, even this high system weight would have to be increased drastically if the system were to be used for urban-vehicle propulsion. Twenty-four hours after shutdown the radiation level is 150 mrem/hr at 24 feet, and during operation, the control cab is kept 500 feet from the reactor. At this distance the radiation is at a safe level. Thus, if the system is to operate at full power, either the control cab must be kept 500 feet away, making it unusable for motor vehicles, or the shielding must be increased drastically.

The approximate dimensions of this system are as follows: reactor system, 9 x 8 x 9 ft; power conversion equipment, 14 x 9 x 8 ft; and control cab, 12 x 7 x 7 ft. As mentioned above, the control cab is located 500 feet from the reactor during operation. It is possible that a more condensed version of the system could be made, but this would increase the radiation problem and, consequently, system weight and cost.

The restart time of the ML-1 system as well as any thermal reactor system following a long shutdown period may be as much as 15 to 20 hours. The reason for this is as follows. During reactor operation, many fission products are formed, two of which are iodine-135 and xenon-135. Xe-135 is a reactor "poison" in that it has a high affinity for absorbing neutrons. Xe-135 is formed from the radioactive decay of I-135. At reactor power for several days, the I-135 and Xe-135 are in equilibrium. However, upon shutdown, the I-135 ceases to be produced but Xe-135 continues to be produced from the decay of I-135 still present. The Xe-135 level rises above its equilibrium level since there are no neutrons present to "burn it out". The peak of Xe-135 occurs about 10 hours after shutdown and most reactors do not incorporate enough fuel to override this "poison" effect to any great extent. It this type of power plant were shut down for a short period, the Xe-135 increase is only slight, so the system could possibly be restarted in 30 to 60 minutes. It takes 60 hp(e) maximum of auxiliary power for start-up. The shutdown time is instantaneous in emergencies. For normal shutdown, however, the auxiliary power system is used and the time required is about 20 to 30 minutes.

diameter and 8 feet in length. If we assume that enough shielding has been supplied, this portion of the system could be attached to the power-conversion portion. The total length of the spacecraft would, therefore, be approximately 26 feet and the diameter approximately 6 feet.

The start-up time for SNAP-8 is 3 hours for manned operations.(104) The shutdown time is instantaneous in emergencies. Normal shutdowns take up to a half hour. This system could probably be redesigned to operate normally with a scram-type shutdown. No appreciable auxiliary power is needed for start-up or shutdown.

The cost of the SNAP-8 type system is very high. NASA estimates the cost of one SNAP-8 system to be 3 to 5 million dollars.(105) This cost estimate does not consider the 110 million dollars already spent in perfecting the SNAP-type system.(106) This price is quite high, but the system is in the experimental stage. With increased production and advances in technology, it is possible that the cost could be brought down by one to two orders of magnitude.

SNAP-50/SPUR System

General Description. The SNAP-50 is being developed jointly by the Atomic Energy Commission and the Air Force. It also uses a closed Rankine-cycle engine to drive an electric alternator. The project has as its aim the demonstration of the feasibility of a space nuclear electric power system rated at 400 to 1340 hp(e) at a specific weight in the neighborhood of 15 lb/hp(e) unshielded.

Although moderators are common in nuclear reactors because they maximize fuel efficiency, they do limit temperature and, hence, overall thermal-cycle efficiency. In addition, moderators are bulky, increasing both the size and weight of a reactor.

For these reasons, it was necessary in SNAP-50 to go to an unmoderated, or fast, design. Absence of a moderator calls for use of fuels with high uranium-235 content. The fuel for SNAP-50 will probably be the oxide, carbide, or nitride of uranium.

The power comparison system for SNAP-50 consists of three loops: the reactor-coolant loop, using lithium; the Rankine-engine loop, using potassium; and the radiator loop, using a sodium-potassium eutectic. In the reactor loop, liquid lithium flows through the reactor core, extracting the heat produced by nuclear fission. Exiting the core at around 2000 F, it flows in a single pipe to a boiler, where some of its heat is transferred to the potassium Rankine-engine loop. In the condenser, the heat lost by the condensing potassium vapor is transferred to the sodium-potassium of the radiator loop.

Physical, Performance, and Cost Characteristics. Most of the technical data and new advances in the SNAP-50/SPUR system are classified. Since this report cannot give up-to-date data on the progress of the SNAP-50, it will be assumed that the design objectives can be reached in the near future, and conclusions will be drawn from these design objectives.

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The weight of the SNAP-50 system, unshielded and with 400 hp(e) output, would be 6000-7000 lb.(107) For urban-vehicle application, however, a man-rated, 4 π shield would be required. As stated previously, the lithium-hydride shield provides the maximum protection with minimum weight. But even this shield at a power between 2700 to 4000 hp(t) which is required to produce 400 hp(e), with a separation distance of 50 feet, and a dose rate of 2.0 mrem/hr, weighs in the neighborhood of 100,000 lb. This results in a specific weight of approximately 250 lb/hp(e). Thus, even though the specific weight of the reactor, power-conversion system, and the radiators has been cut down by the SNAP-50-type system, the weight of the shielding still makes it unacceptable for an urban-vehicle propulsion system.

The SNAP-50 power plant is 10 feet in diameter and about 35 feet long.(108) This is assuming enough shielding can be provided so the passenger section can be connected to the power-conversion section.

The start-up time for the SNAP-50 is approximately 3 hours. This time is required to develop the operating conditions in the external loops. The shutdown time is instantaneous. Neither start-up nor shutdown require any extensive auxiliary power source.

Since the SNAP-50 system is still in the development stage, there are no cost figures. It is probable, however, that development of this system will cost around 2 billion dollars, and it is improbable that the cost of one system could be reduced to the $5,000 to $25,000 range.

Summary of Characteristics and Conclusion as to Feasibility

Three main types of portable reactor systems have been investigated. These include (1) the ML-1, a water-moderated system with a water and lead shield; (2) SNAP-8, a uranium-fueled and zirconium hydride-moderated reactor with a lithium hydride shield; and (3) SNAP-50/SPUR, a fast reactor (no moderator) with a lithium hydride shield. Water-cooled reactor systems were not investigated because by their nature they are not portable

The cost of any of these reactor systems is in the millions of dollars. The specific weight of the most compact, completely shielded system would be greater than 250 lb/hp(e) or hp(s). The start-up time of all three systems is impractical.

It may be feasible in the future to lower the cost of materials and to devise some system to overcome the long start-up time. It is unlikely that the shielding weight problem will be solved, however. The radiation near a power reactor is so intense that it takes enormous amounts of shielding to provide maximum protection with the minimum weight. A hundredfold improvement in the protection-per-unit-weight characteristic of shielding materials is required before nuclear reactors would be suitable for vehicle use. For these reasons, nuclear reactor systems must be ruled out as potential propulsion systems for urban vehicles.

Radioisotope Thermal Power Generators

Since the advent of nuclear-energy research, radioisotopes have become increasingly important in a variety of applications. One of these involves the use of radioisotopes as the energy source for power generators. The energy released by radioisotopes may be converted either into thermal energy, which may then be used as the energy input to a variety of heat engines and direct thermal-to-electric energy converters, or directly into electrical energy. Radioisotope thermal power generators are discussed in this section of the report and direct nuclear-to-electric power generators are discussed in the following section.

The radiation emitted by radioisotopes is mainly of three types, alpha, beta, and gamma. An alpha particle is identical with the nucleus of helium atom. It has very little penetrating power. Beta particles, usually negative electrons, are much more penetrating but less damaging than alpha particles. Gamma rays are highly penetrating electromagnetic waves similar to X-rays.

When nuclear particles or gamma rays are absorbed in a thermal generator, they produce highly localized heating effects along their trajectories (thermal spikes). The effects almost instantaneously average out, and heat is produced on a macroscopic scale.

Specific Power of Generator

An important parameter in radioisotope generator design is the specific power, P_{sp}, of the fuel in w/g. Once the average amount of energy absorbed for each disintegration within the generator, E_{av}, is determined, a simple equation for P_{sp} may be derived. It is:

$$P_{sp} = 2.12 \times 10^3 \, RE_{av}/AT_{1/2} \quad ,$$

where

P_{sp} = specific power, w/g of fuel compound

E_{av} = the average energy absorbed in the fuel per disintergration, Mev

$T_{1/2}$ = half life, yr

A = atomic mass, g/mole

R = mass of the isotope per unit mass of fuel compound.

This equation is valid only for a pure isotope at time (t) = 0. It can be multiplied by $e^{-\lambda t}$ to bring in time dependence with λ being a characteristic factor for an individual isotype. It is desirable to have both $T_{1/2}$ and P_{sp} large, but as the equation shows, they are reciprocally related.

General Engineering Considerations

Proceeding to the purely engineering matters involved in the construction and the operation of radioisotope power generators, it is noted that radioisotopes do not require the formation of a critical assembly as fission reactors do. In principle, radioisotopes could be used in any quantity, in any configuration, to attain any power level desired. However, there are technical problems in application.

First, the fuel must be shielded. This leads to consideration of a compact fuel body. Thus, the designer is faced with the problem of extracting heat from a compact structure. At high power levels (tens of horsepower thermal power), holes must be fabricated in the fuel block and the thermal energy transferred by fluid convection since pure heat conduction and radiation are no longer adequate. This approach leads to moving parts (pumps) and thus the reliability of the system is reduced.

The second and most serious roadblock to the attainment of high power levels with radioisotopes is the scarcity of fuel. Table 23 (page 129) shows the quantities of radioisotopes that will probably be available over the next decade. The projected power rating for all of the usefull radioisotopes produced per year (assuming a nominal conversion efficiency of 10 percent) is only about 2000 hp(e)/yr over the 1970-1980 period. This is not much power by terrestrial standards. The concensus is for conserving the limited supply of radioisotopes for use in generators in a low-power range rather than attempting to construct multihosepower-sized power plants that are better served by fission reactors. Radioisotopes can perform at these low-power levels where other power sources are totally unacceptable. But if radioisotopes are to be used in horsepower size power systems their availability is of major importance. Therefore, the availability of radioisotopes is critical in their application for motor vehicles.

Another engineering constraint is imposed by a basic property of radioisotopes, the exponential decay of their energy output with time. As the isotope decays, so does the flux and the thermal-energy-reservoir temperature. There are three ways to combat this problem: (1) store the excess energy at the beginning of life and use it later to boost the power to the required level; (2) employ a higher-than-necessary thermal power at the beginning of life, discarding the excess energy, and terminate the use of the heat source when the power has dropped to the design level; (3) use a radioisotope whose daughter also generates power; and (4) employ long-lived fuels. All generators now in operation have used the last approach in solving the power-flattening problem by using seed fuels like Pu^{238} and Sr^{90}.

The inexorable decay of the radioisotope fuels results in a generator that cannot be turned on or off at will. Thermal power will be produced continuously, although electrical energy may be diverted or controlled if desired. Hence, cooling provisions must be provided to remove the waste heat from the generator at every step in the installation and operation. This is a serious restriction of the use of radioisotopes in systems with intermittent power demand.

Selection of Radioisotope Fuels

In selecting the best radioisotope for power production, high power density, long life and low cost are desirable in a potential fuel. Conversely, fuels of high toxicity or those emitting high energy gamma rays are undesirable. A systematic selection procedure must be set up to choose the few best among the more than 1,000 possible radioisotopes.

The first sifting is done on the basis of physical properties. Using the following criteria, the many possibilities can be reduced to 36.

(1) The half life should be greater than 100 days and less than 100 years. The 100-day limitation eliminates the many short-lived isotopes that would present severe fuel processing and power-flatening problems. Fuel should have a half life of at least a few months to account for encapsulation time and to permit some stockpiling. Radioisotopes with half lives over 100 years generally have unacceptably low specific powers, as shown by the equation on page 126.

(2) The specific power should be greater than 0.06 hp(t)/lb. This criteria is established to eliminate the many nuclei that emit a few weak particles but still have acceptable half lives. The larger fuel capsules associated with low specific power fuels lead to higher shield weight and lower generator efficiencies.

(3) Pure or nearly pure gamma emitters should be eliminated because of their shielding and handling problems.

The first sieve just described is too coarse for practical purposes, so a second screening is performed using chemical and engineering properties for guidelines. The two criteria in this second sieve are:

(1) The fuel should be relatively noncorrosive, compatible with structural materials, and stable in time.

(2) The fuel should have good engineering properties at moderately high temperatures. The engineering properties of interest are the practical power density, melting point, dimensional stability, gas evolution (e.g., helium buildup in alpha emitter), thermal conductivity, and density.

Only 17 potential fuels remain after these criteria are applied. From this group the Atomic Energy Commission has focused its production efforts on eight radioisotopes. These fuels, listed in Table 23, have the most appropriate properties for vehicle application.

The first four isotopes in Table 23 – all beta emitters – are fission products that are recovered from nuclear-fuel-reprocessing plants. The last four, which emit alpha particles, must be produced artificially in nuclear reactors. The alpha emitters are currently more expensive than the fission products, but they are much easier to shield and have good power densities. For use as an automobile power source, the higher cost of alpha-emitting fuel is more than offset by the weight reduction because heavy radiation shields are not needed.

TABLE 23. CHARACTERISTICS OF RADIOISOTOPIC HEAT SOURCES(109, 110)

	Sr-90	Cs-137	Ce-144	Pm-147	Po-210	Pu-238	Cm-242	Cm-244
1. Half Life, yr	27.7	26.6	0.78	2.62	0.38	86.4	0.45	18.10
2. Compound Form	SrO	CsCl	CeO_2	Pm_2O_3	Metal	PuO_2	Cm_2O_3	Cm_2O_3
3. Compound Specific Power, hp(t)/lb	0.48	0.13	13.00	0.19	86.00	0.30	67.00	1.70
4. Power Density, Compound, hp(t)/ft(3)	140	33.0	5,800	77.3	49,300	212	44,800	1,140
5. Types of Radiation, Major(a)	$\beta\chi$	$\beta\gamma\chi$	$\beta\gamma\chi$	$\beta\gamma\chi$	α	αn	αn	αn
6. Shielding Required	Heavy	Heavy	Heavy	Minor	Minor	Minor	Minor	Minor
7. Melting Point, F	4,400	1,195	4,856	2,732	489	4,082	>2,732	>2,732
8. Annual Availability, 1967-70, hp(t) Annual	120	64	107	7	12,600	20	160	134
9. Present Price, $/hp(t)	22,000	19,000	14,000	420,000	580,000	1,200,000	123,000	324,000
10. Projected Annual Availability, 1970-80(c), hp(t) Annual	870	830	5,600	75	12,600	80	NA	NA
11. Minimum Estimated cost, Encapsulated, 1970-80, $/hp(t)	15,000	19,000	1,500	70,000	7,500	400,000	NA	NA

(a) Legend: α- Alpha, β-Beta, γ-Gamma, n-Neutron, χ-Penetrating bremsstrahlung.

(b) NA-not available.

(c) AEC Division of Isotope Development projections as of July 1965. These have undoubtedly been increased in more recent projections.

Assuming a 13.4 hp(e) source [about 67 hp(t)] with a separation distance of 3 feet, and 0.5 rem/hr radiation dose allowed, the shielding weight for Sr-90, Cs-137, and Ce-144 would exceed 5000 lb. Thus, these three radioisotopes are not suitable for urban-vehicle applications.

The melting point for Po-210 (489 F) is unacceptably low. This results from the fact that energy converters used in vehicle applications will reject heat to the ambient air. Allowing suitable temperature differentials for heat transfer on both the hot and cold sides of the energy converter plus some safety factor with respect to the melting temperature of Po-210 would probably result in an effective temperature difference of less than 200 F across the energy converter. This would result in an unacceptably low efficiency and large energy converter. Had it not been for its low melting temperature Po-210 would have been the preferred isotope for vehicle application.

Of the remaining four isotopes, Cm-242 would appear to be most appropriate for automotive application. It currently has the lowest price of the four and its specific power in compound form is quite high.* Unfortunately, even though a vehicle propulsion system using Cm-242 as the energy source might be developed to have acceptable physical and performance characteristics, the ultrahigh cost and unavailability of Cm-242, or any of the other radioisotopes for that matter, makes the use of radioisotopes unfeasible for vehicle application in the foreseeable future.

Summary of Characteristics and Conclusion as to Feasibility

Radioisotope thermal power generators could probably be developed to have weight and size characteristics that would be acceptable for vehicle application. They have a number of overriding shortcomings, however, that make them unfeasible now and in the foreseeable future for urban-vehicle application. The first of these are their very high cost and unavailability. The isotope identified as most appropriate for vehicle application, CM-242, currently costs \$123,000/hp(t) and its current annual availability is only 160 hp(t). While other isotopes could be considerably less costly and are more readily available than Cm-242, particularly Po-210, they have other drawbacks that make them undesirable for vehicle application. In any case, the tremendous cost reduction and increase in availability required to make radioisotopes feasible for vehicle application appear to be far in the future.

Another major problem with radioisotopes is their inexorable decay which results in a thermal generator that cannot be "turned off". If a radioisotope thermal generator of "cruise power" size were used, a large increase in the "thermal pollution" of the city would result. The use of low power radioisotope-thermal generators combined with thermal energy storage (discussed earlier in this report) would not be as desirable as using electric or methane heaters for this usage.

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*The specific power of a radioisotope thermal-energy source would be much lower than that of the radioisotope compound alone, as to remove the heat generated by the radioisotope, a suitable heat exchanger along with auxiliaries such as pumps would be required. The radioisotope could then be deposited on one side of the surface of this heat exchanger.

Radioisotope Direct Electric Power Generators

Two methods for converting nuclear energy from a radioisotope source directly into electric energy were investigated: (1) collection of charged particles in a retarding field, and (2) generation of ion pairs in the neighborhood of a solid-state p-n junction. The alpha cell is a typical example of the first, and the betavoltaic cell the second.

The Alpha Cell

General Description. The alpha particle is essentially a helium atom with the two electrons removed. Certain radioisotopes decay by the emission of an alpha particle, the half-life of the decay varying widely with the isotope. The alpha particles emitted are monoenergetic, with energies for isotopes of interest typically in the range of 5 to 6 million electron volts (Mev). At birth each alpha particle has a charge of +2. Thus, the underlying principle of the alpha cell process makes use of the fact that this alpha decay process yields charged particles in motion.

If an alpha emitting isotope is distributed in a sufficiently thin layer, rather than a thick fuel region, so that an appreciable fraction of all the alpha particles produced in the layer can escape from the surface with much of their initial energy and charge intact, these particles can be collected on an insulated electrode. The first few alpha particles reaching the electrode will deposit their charge and dissipate their energy as heat. However, after a number of alphas have been collected, the insulated electrode, by virtue of its surplus of positive charge, will attain a high voltage with respect to the emitter layer. Subsequent alpha particles will do work against this field: they will arrive at the electrode with their initial kinetic energy exhausted but will deposit the charge.

In effect, the arrangement discussed above is analogous to a capacitor, as shown in Figure 29, with the alpha particles doing the charging. The charge separation caused by the energetic alpha particles driving their way to the insulated electrode, the collector, is neutralized by a flow through an external circuit of the electrons which were left behind. This electron flow through an external circuit is a source of direct electricity produced without the use of a heat cycle.

The discovery that charged-particle emission can buildup a voltage on a properly insulated electrode may be traced to work by Mosley in 1913. A critical problem existed in reducing the concept to practice. Along with the positively charged alpha particles come secondary electrons which have an opposing charge. Measurements indicate that approximately 10 secondary electrons are released from the surface with each alpha particle.(111) Although these secondary electrons have very low energy (approximately 97 percent have energy less than 100 ev), they are produced in such abundance that their total negative charge more than offsets the positive charge buildup. Thus, a low-energy negative charge, rather than the desired positive high-energy charge, is emitted from the cathode.

A properly designed control grid placed close to the cathode can be used to overcome this difficulty. A negative potential applied to the grid will repel the secondary electrons to the cathode. A simple configuration which illustrates this concept is shown

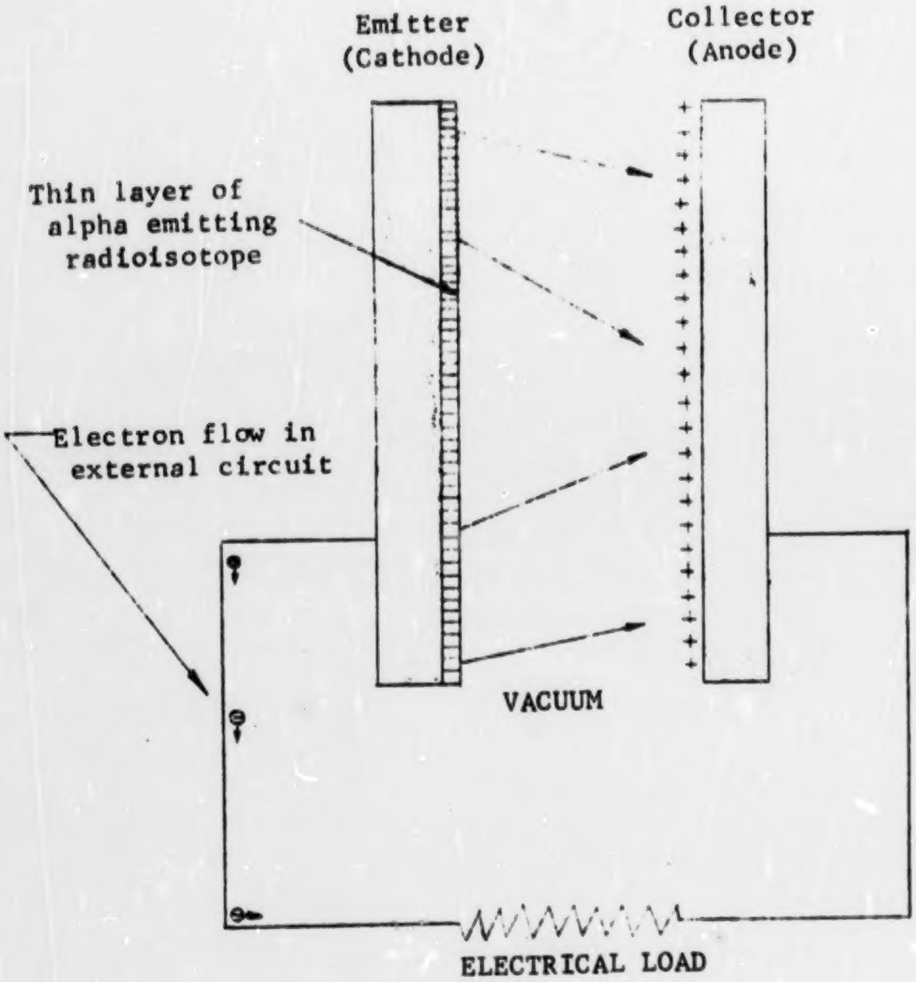
Emitter
(Cathode)
Collector
(Anode)
Thin layer of alpha emitting radioisotope
Electron flow in external circuit
VACUUM
ELECTRICAL LOAD

FIGURE 29. THE PRINCIPLE OF THE ALPHA CELL'S OPERATION

in Figure 30. A cylinder electrode, the emitter or cathode, can be a tube or rod coated with a layer of alpha emitter several tenths of thousandths of an inch thick. This thickness is required since the range of alpha particles in metals is of the order of 4×10^{-4} inch. Surrounding the cathode in this example is a squirrel-cage grid at negative voltage with respect to the cathode. The grid is composed of small-diameter wires and is sufficiently open to permit the alpha particles to reach the collector, or anode, while at the same time preventing the secondary electrons from escaping. Because the electrons have very low energy, only a few hundred volts bias on the grid will suppress them. To hold back the electrons when the anode is at several million volts requires an increase in voltage to the kilovolt range. The grid can be designed to be almost completely open, typically 90 percent or more of the total grid area is not blocked by grid wires.

Physical, Performance, and Cost Characteristics. Experiments were performed in late 1962 which showed that the principles of operation of the device were sound in the relatively low voltage range obtained at that time (approximately 50,000 volts). However, for efficient operation it has been calculated that the device must operate in the range near 1 megavolt. Because of microdischarging in the cell, significantly higher voltage cannot be obtained at this time. Because of these microdischarging problems, present efficiency achieved with alpha cells is less than 5 percent.

The power range for a generator based on the alpha-cell direct-conversion concept is not yet firmly established. However, it appears at present that individual generator cells with useful outputs of 0.013 to 0.13 hp(e) may be achieved.

Since the radioisotope used in the alpha cell must be spread on a tube or rod in a layer approximately 4×10^{-4} inch thick and since low energy conversion efficiencies are achieved, surface areas of thousands of square feet would be required for several tens of horsepower electrical output.

As with all radioisotope systems, a major problem with an alpha cell system is the cost and availability of the fuel. The most promising alpha emitters are Pu-238, Cm-242, and Cm-244. For use in an alpha cell, the least expensive of these, Cm-242, would cost $123,000/hp(t).

Betavoltaic Devices

General Description. When beta particles are absorbed by a semiconductor they dissipate most of their energy by ionizing the atoms of the solid. The carriers generated in this fashion are in excess of the number permitted by thermodynamic equilibrium and if they diffuse to the vicinity of a rectifying junction they induce a voltage across the junction. This phenomenon, which has been termed the electron-voltaic effect is the basis of the beta voltaic cell.

This effect has been studied as a possible source of electrical power since it permits the direct conversion of the energy of beta particles emitted by a radioactive material into electricity.(113)

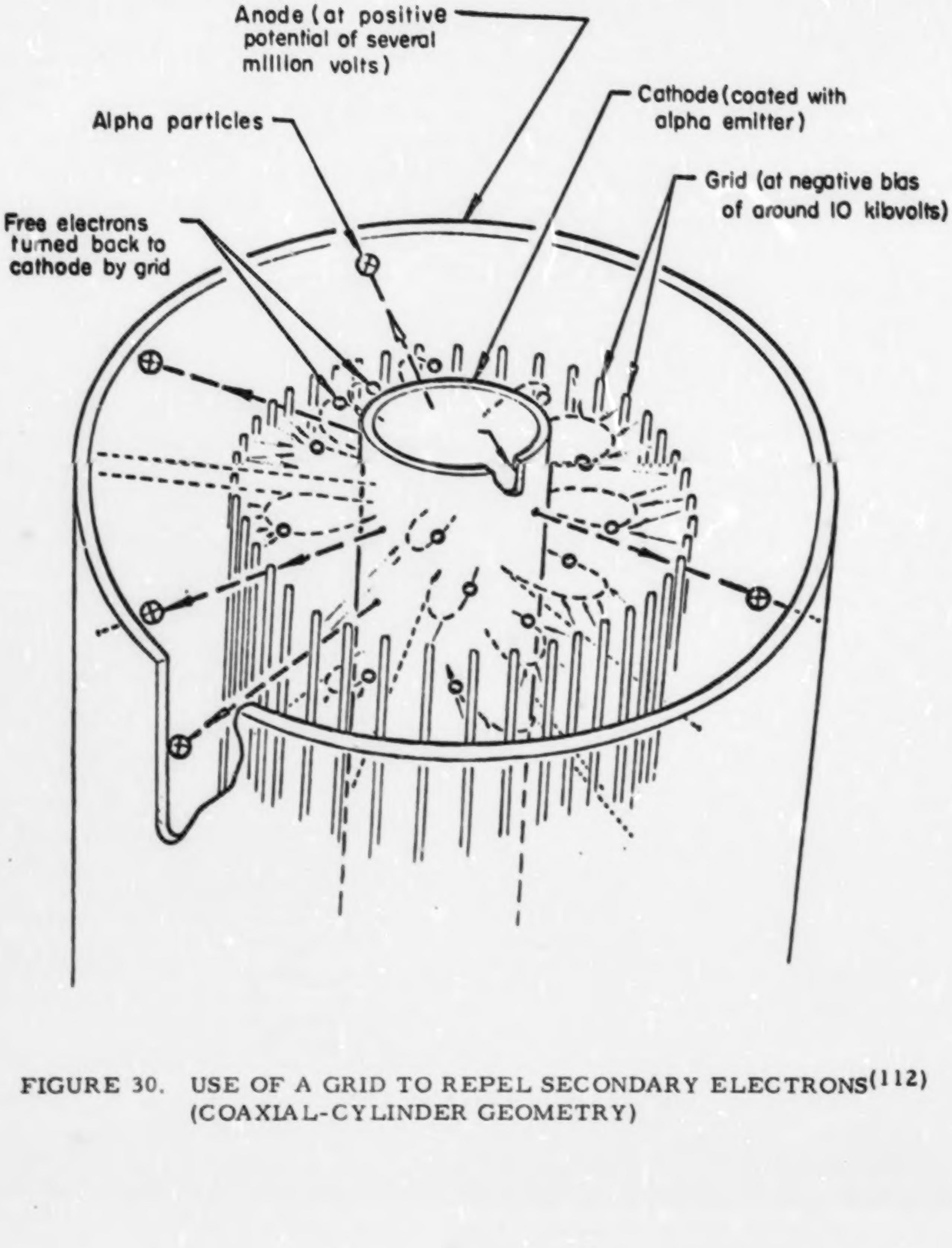
Anode (at positive potential of several million volts)
Cathode (coated with alpha emitter)
Alpha particles
Grid (at negative bias of around 10 kilovolts)
Free electrons turned back to cathode by grid

FIGURE 30. USE OF A GRID TO REPEL SECONDARY ELECTRONS(112) (COAXIAL-CYLINDER GEOMETRY)

The electron-voltaic effect is one of a class of phenomena (the best known is the photoelectric effect) in which radiation capable of producing ionization in a solid interacts with a rectifying junction. Other forms of electromagnetic radiation (γ-rays and x-rays) as well as other forms of radiation (β particles, protons) are possible sources of such ionization, and the analysis of this effect could be applied to these other types of radiation as well. Beta rays were chosen as the primary source for direct conversion because their specific ionization is higher than that of γ-rays. They are, therefore, absorbed in a reasonable thickness of semiconductor and they can be shielded more easily. They were preferred over α particles because they are emitted by radioisotope by-products of the fission process and because their smaller weight produces less radiation damage.

When determining which beta emitter to use as the radioactive source, the radiation damage caused by the source must be considered. It was found that the power output of the cells decreased rapidly because of radiation damage caused by the high-energy (up to 2 Mev) electrons emitted by a Sr-90 radioisotope used as the primary power source. Subsequent work has shown that the energy of bombarding electrons had to exceed a few hundred Kev in order to produce the observed degradation in power output. Thus, a radioisotope must be chosen which emits primarily low energy beta particles. For this reason Pm-147 is commonly used. Its half life is shorter than that of Sr-90, but the maximum energy of the emitted β-particles is only 230 kev.

Physical, Performance, and Cost Characteristics. Each betavoltaic cell consists of a thin layer of Pm-147 in the form of hydrated Pm_2O_3, sandwiched between two n-on-p silicon solar cells and encapsulated in a stainless steel case. It is approximately 1 cu in. in volume. The most economical thickness of the Pm-147 would be about 4×10^{-4} inch. If the thickness is increased beyond this, the power would not increase significantly.

In a scoping experiment (all factors not optimized), a 2-curie Pm-147 source generated about 1.0×10^{-6} hp (beta power). The amount of energy transferred from a Pm-147 source to one cell is reduced by self-absorption in the source and, to a lesser extent, back-scattering from the cell. The maximum theoretical cell efficiency is approximately 4.4 percent. Self-absorption in the 2-curie sources involved in the present batteries probably reduce the power input to the cells by 75 percent, which would indicate a maximum efficiency of about 1.1 percent for electrical power generation from beta particles entering the cells and a maximum overall efficiency of about 0.8 percent. It is rather obvious that significant improvements in efficiency will have to be made before betavoltaic devices can be used in a large power system.

Assuming that present betavoltaic cells are about 1 cu in. in volume, have an efficiency of 1.0 percent, and have a 1.0×10^{-6} hp (beta power) source, 100 million cells would be required to produce an output of 1 hp(e). If these were assembled into one package, it would be about 40 feet on a side. The weight of such a betavoltaic device including cell connectors and appropriate shielding would be greater than 10,000 lb.

The cost of a horsepower-size betavoltaic system would be astronomical. The cost of the fuel, Pm-147, if it could be obtained, would be in the millions of dollars. The projected cost of Pm-147 for the 1970-1980 period is $70,000 hp(t). It is not likely that advances in technology will enable the cost to be lowered to an acceptable level in the intermediate future.

Summary of Characteristics and Conclusions as to Feasibility

The excessive cost, unavailability, and "no-turn-off" problems cited for radioisotope thermal power generators apply to radioisotope direct electric power generators, as well. In addition, however, the direct electric power generators have some problems of their own.

The alpha cell requires considerable development to even achiev practicality at very low power levels. Operation in the horsepower range cannot be expected in the near future. The problem of microdischarging must be solved before the system can be further examined for practicality.

The power output of a betavoltaic cell is limited by its low efficiency, currently 0.8 percent, and because only a very thin layer of radioisotope can be used per cell. High power levels are unlikely because of the weight and size of the system. Currently, 100 million cells would be required for a 1.0 hp(e) output. This would require a volume 40 feet on a side and would weight more than 10,000 lb.

For the above reasons, radioisotope direct electric power generators are unfeasible now and in the forseeable future for urban-vehicle application.

APPENDIXES

APPENDIX A – DEVELOPMENT OF PROPULSION-SYSTEM VOLUME AND COST SPECIFICATIONS

APPENDIX B – CALCULATION SHEETS FOR THE ESTIMATION OF PROPULSION-SYSTEM CHARACTERISTICS FOR VARIOUS VEHICLE APPLICATIONS

APPENDIX C – REFERENCES

APPENDIX A

DEVELOPMENT OF PROPULSION-SYSTEM VOLUME AND COST SPECIFICATIONS

APPENDIX A

DEVELOPMENT OF PROPULSION-SYSTEM VOLUME AND COST SPECIFICATIONS

Introduction

The open literature contains a number of vehicle-criteria analyses based on permissible component weight, but similar detailed information involving the parameters of volume and cost could not be found. In view of this critical data gap, it became necessary to conduct a cursory cost and volume evaluation for a range of current American vehicles. The vehicles selected for this analysis included four passenger cars, two delivery vans, and one city bus.

The volume data sought were the envelope volumes for the particular existing systems, and the cost data of interest were the manufacturers' minimum selling prices (i.e., O.E.M. or dealer's cost, manufacturer's plant). Since the purpose of this analysis was to establish acceptable limits or threshold values for volume and cost, it was decided to consider each vehicle as equipped with its largest optional engine and transmission. In each of the passenger cars, therefore, a V-8 engine (nonracing version equipped with a 4-barrel carburetor and dual exhausts) and a three-speed automatic transmission were considered. In the case of the delivery vans, both makes offered a three-speed automatic transmission, one did not offer a V-8 engine option, and neither had dual exhausts. The city bus evaluated was equipped with a V-6 diesel engine, an automatic transmission, and a single exhaust system. The procedures used in this analysis, and the results, follow.

Volume-Estimation Procedure

The determination of individual component volumes consisted essentially of deriving envelope dimensions from product literature and/or direct measurement on sample vehicles. In instances where pertinent product literature was not available and the component was inaccessible for measurement, estimates were made by scaling from known similar components.

The term "envelope volume" as used in this discussion refers to the volume of a rectangular and/or cylindrical container which would hold the component being considered and provide a slight operating clearance with adjacent chassis or body structures. The use of envelope volumes and largest optional components was selected as the best means of developing liberal volume limits for the alternative propulsion systems. The conservatism of these limits is increased by permitting the lower performance systems being considered in this program to occupy the maximum allotted volumes for present, high-performance systems. In this study, the total propulsion system volume is defined as the sum of five different component volumes - engine, transmission, driveline, exhaust system, and fuel tank.

Engine Envelope Volume

The engine envelope was taken as that rectangular space enclosed by the longitudinal distance spanning the rear face of the engine cylinder block and the forward edge of the radiator; the maximum width dimension across the engine exhaust manifolds and/or accessories; and the vertical distance spanning the top of the air cleaner and the bottom of the crankcase oil pan. While much of this space is actually vacant, the bulk of the vacancy is required for cooling, isolation, and servicing. The battery and leads are not normally located within the rectangular space just described, but were assumed to be so located in this analysis.

Transmission Envelope Volume

The transmission envelope was chosen to consist of one cylindrical disk which would enclose the torque converter section of the transmission case, and then one or two cylindrical or rectangular containers which would hold the gear box and rear-bearing retainer portions. It should be noted that automatic transmissions occupy slightly more space than do the corresponding manual transmission and clutch combinations.

Fuel Tank Envelope Volume

Since fuel tanks of the subject vehicles were all fairly regular in shape, tank envelope volume was based on the space required to hold the nominal fuel capacity plus 10 percent.

Driveline Envelope Volume

The term "driveline" as used here comprises both the propeller shaft and the rear axle assembly. The propeller shaft envelope volume was taken as the area required by the universal joints times the length of the shaft. Volume assigned to the rear axle assembly consisted of a cylindrical disk which would enclose the differential section of the axle housing and two tubular spaces which would hold the axle-shaft portions of the housing. No arbitrary correction was made to these figures to account for maximum vertical displacement of the driveline with respect to the body.

Exhaust-System Envelope Volume

Exhaust-system envelope volume was based on cross-sectional pipe and muffler areas slightly larger than normal, scaled muffler length, and an estimated developed length for the pipe.

Cost-Estimation Procedure

While several consumer buying guides such as the American Car Price Magazine list "manufacturing costs" for complete vehicles, similar information for vehicle components is, of course, not available. All of the cost data presented here, therefore, are estimates based on suggested retail costs for vehicles, components, and optional equipment; typical dealer and employer discounts; retail prices of rebuilt units; acceptable overhaul expenses; and the specific cost for similar equipment.

In general, the final estimated cost fell between the suggested retail price (minus typical OEM discount and shipping charges) and the suggested retail price of a rebuilt unit (minus typical discount and shipping charges). However, in view of the gross inconsistencies discovered in pricing policies, no simple estimating procedure was found suitable and each component had to be evaluated on the basis of all available inputs.

Again, a degree of conservatism is introduced by permitting the low-performance vehicles being considered in this program to have the same propulsion-system cost as current high-performance vehicles. This degree of conservatism does not exist in the case of the bus propulsion system, however, so the permissible bus propulsion-system cost was increased 30 percent above the current estimated cost.

Results and Conclusions

The principal findings of this analysis are as follows:

(1) For passenger cars, propulsion-system volume and cost are related to vehicle curb weight as indicated in Figures A-1 and A-2. Choosing the extreme cases and extrapolating yields the limit specifications tabulated below.

	Family Car	Commuter Car	Utility Car	City Taxi
Curb Weight, lb	3500	2100	1400	3500
Propulsion System Volume, ft^3	28	20	16	28
Propulsion System Cost, $	820	670	600	820

(2) For delivery vans with a curb weight of 4500 lb and a GVW rating of 7000 lb, the average propulsion-system volume and cost were estimated to be 28 ft^3 and $860, respectively.

(3) For a city bus with a curb weight of 20,000 lb and a GVW rating of 30,000 lb, the current propulsion-system volume and cost were estimated to be 134 ft^3 and $6,240, respectively. Since the maximum power specified for the low-pollutant bus is similar to that of present conventional buses, permissible volume and cost for a low-pollutant system will be increased 30 percent over the current estimated values.

(4) Scatter observed in the passenger-car volume data is mainly a function of radiator and air-cleaner placement relative to the engine.

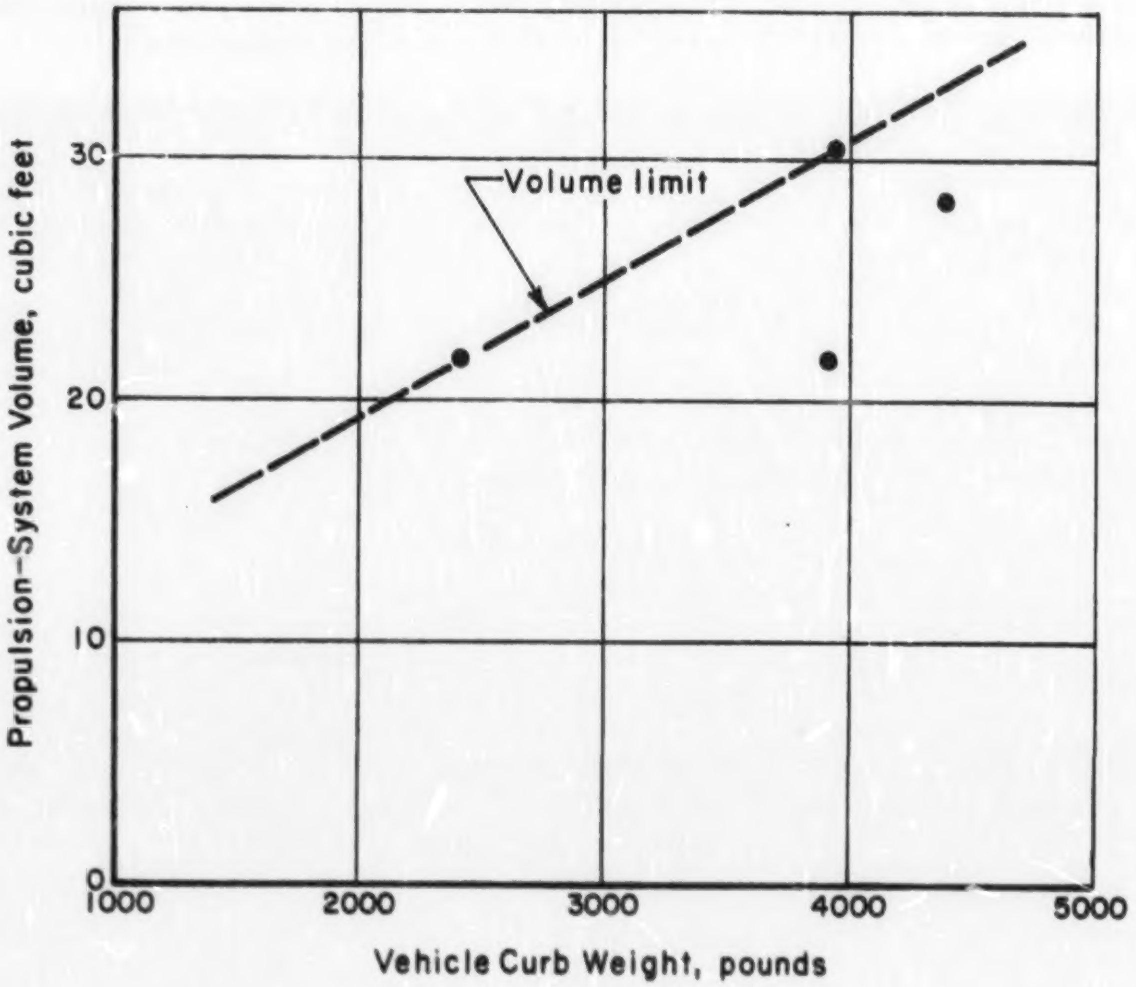

FIGURE A-1. PROPULSION SYSTEM VOLUME VS VEHICLE CURB WEIGHT

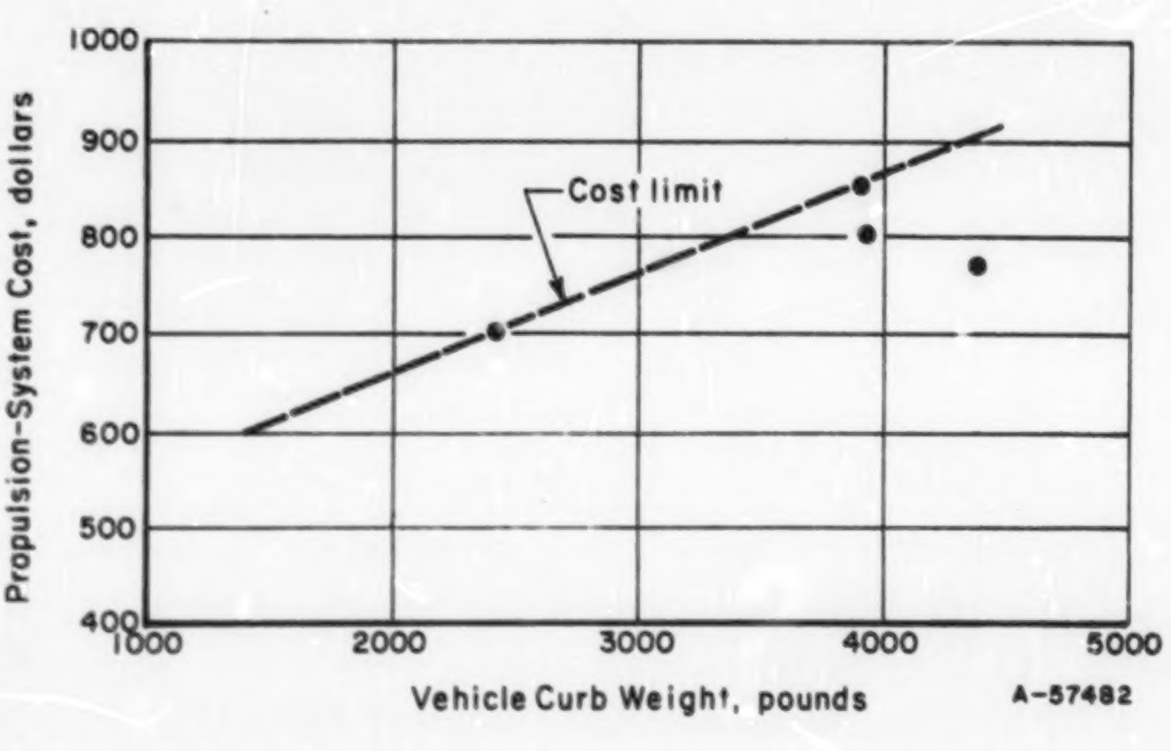

FIGURE A-2. PROPULSION SYSTEM COST VS VEHICLE CURB WEIGHT

(5) Limiting propulsion-system volume and cost specifications for passenger cars and light-duty delivery vans are nearly the same because many of the components are identical.

(6) While the rated horsepower of the city bus is less than that of the smallest passenger car considered, its volume and cost specifications are grossly higher because of its heavy-duty components and low volume production.

APPENDIX B

CALCULATION SHEETS FOR THE ESTIMATION OF PROPULSION-SYSTEM CHARACTERISTICS FOR VARIOUS VEHICLE APPLICATIONS

TABLE B-1. ESTIMATED PROPULSION SYSTEM CHARACTERISTICS FOR COMMUTER CAR APPLICATION

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter[a]	Thermionic Converter[a]	Thermo-photo-voltaic Converter[a]	Magneto-hydrodynamics and Nuclear Devices
Maximum Delivered Power, hp(s)	30	30	30	30	30	30	30	Found to be unsuitable for this application – see discussion.
Assumed Transmission and Drive Efficiency, %	90	95	85	80	80	80	80	
Maximum Engine Power, hp(s) or hp(e)	32.4	31.6	35.4	37.5	37.5	37.5	37.5	
Rated Engine Power, hp(s) or hp(e)	32.4	25.2	35.4	37.5	19	19	19	
Specific Engine Weight, lb/hp(s) or hp(e)	4.0	7.0	9.4	--	13	10	13	
Engine Weight, lb	130	180	330	--	250	190	250	
Specific Transmission Weight, lb/hp(s)	0.5	0.0	0.7	2.0	--	--	--	
Transmission Weight, lb	17	0.0	24	75	Weight of battery, motors, and controls			
Energy Delivered to Vehicle, hp(s)-hr	13.4	13.4	13.4	13.4	13.4	13.4	13.4	
Energy Delivered by Engine, hp(s)-hr or hp(e)-hr	14.9	14.1	15.8	--	16.8	16.8	16.8	
Peak Engine Efficiency, %	30	26	36	--	10	10	13	
Assumed Average Efficiency, %	21	18	25	80	9	9	11.5	
Required Stored Energy, hp(t)-hr	71	78.4	63.2	16.8[b]	187	187	146	
Specific Weight of Conventional Fuel, lb/hp(t)-hr	0.15	0.15	0.15	100[c]	0.15	0.15	0.15	
Energy Storage Weight, lb	11	12	10	1680	28	28	22	
Engine, Transmission, and Conventional Fuel Weight, lb	160	190	360	1800	280+	220+	270+	
Specific Engine Volume, ft^3/hp(s) or hp(e)	0.12	0.15	0.18	--	--	0.2	--	
Engine Volume, ft^3	3.89	3.78	6.39	--	--	3.80	--	
Specific Transmission Volume, ft^3/hp(s)	0.004	0.0	0.005	0.02	--	--	--	
Transmission Volume, ft^3	0.13	0.0	0.18	0.75	Volume of battery, motors, and controls			
Specific Volume of Conventional Fuel, ft^3/hp(t)-hr	0.0029	0.0029	0.0029	0.25[d]	0.0029	0.0029	0.0029	

TABLE B-1. (Continued)

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter[a]	Thermionic Converter[a]	Thermo-photo-voltaic Converter[a]	Magneto-hydrodynamics and Nuclear Devices
Energy Storage Volume, ft^3	0.21	0.23	0.18	4.20	0.54	0.54	0.42	Found to be unsuitable for this application – see discussion.
Engine, Transmission, and Conventional Fuel Volume, ft^3	4.2	4.0	6.7	4.9	Similar to Thermionic	4.3^+	Similar to Thermionic	
Specific Engine Cost, \$/hp(s) or hp(e)	4.5	5.4	6.4	--	Not known but is expected to be \$100/hp(e) or higher			
Engine Cost, \$	150	140	230	--	--	--	--	
Specific Transmission Cost, \$/hp(s)	0.4	0.0	1.0	3.0	--	--	--	
Transmission Cost, \$	13	0.0	35	110	Cost of batteries, motors, and controls			
Specific Energy Container Cost, \$/hp(t)-hr	Nominal	Nominal	Nominal	100[e]	Nominal	Nominal	Nominal	
Energy Container Cost, \$	--	--	--	1680	--	--	--	
Engine, Transmission, and Energy Container Cost, \$	160	140	260	1800	Very high			

(a) Considered to be part of a hybrid system. Their power rating is based on the cruise power requirement which is one-half the maximum power. The weight, size, and cost of associated batteries, motors, and controls would have to be added to the values listed in this table for the converters alone.

(b) Value given in hp(s)-hr.

(c) Value given in lb/hp(s)-hr.

(d) Value given in ft^3/hp(s)-hr.

(e) Value given in \$/hp(s)-hr.

TABLE B-2. ESTIMATED PROPULSION-SYSTEM CHARACTERISTICS FOR UTILITY CAR APPLICATION

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter[(a)]	Thermionic Converter[(a)]	Thermo-photo-voltaic Converter[(a)]	Magneto-hydrodynamics and Nuclear Devices
Maximum Delivered Power, hp(s)	It is not believed that a competitive vehicular gas turbine of this small size could be developed.	16	16	16	16	16	16	Found to be unsuitable for this application - see discussion.
Assumed Transmission and Drive Efficiency, %		95	85	80	80	80	80	
Maximum Engine Power, hp(s) or hp(e)		16.9	18.8	20	20	20	20	
Rated Engine Power, hp(s) or hp(e)		13.5	18.8	20	10	10	10	
Specific Engine Weight, lb/hp(s) or hp(e)		8.0	10.0	--	13	10	13	
Engine Weight, lb		110	190	--	130	100	130	
Specific Transmission Weight, lb/hp(s)		0.0	0.7	2.0	--	--	--	
Transmission Weight, lb		0.0	13	40	Weight of battery, motors, and controls			
Energy Delivered to Vehicle, hp(s)-hr		10.7	10.7	10.7	10.7	10.7	10.7	
Energy Delivered by Engine, hp(s)-hr or hp(e)-hr		11.3	12.4	--	13.4	13.4	13.4	
Peak Engine Efficiency, %		25	35	--	10	10	13	
Assumed Average Efficiency, %		17.5	24.5	80	9	9	11.5	
Required Stored Energy, hp(t)-hr		64.6	50.6	13.4[(b)]	149	149	116	
Specific Weight of Conventional Fuel, lb/hp(t)-hr		0.15	0.15	100[(c)]	0.15	0.15	0.15	
Energy Storage Weight, lb		10	8	1340	22	22	18	
Engine, Transmission, and Conventional Fuel Weight, lb	--	120	210	1400	150+	120+	150+	
Specific Engine Volume, ft^3/hp(s) or hp(e)		0.16	0.20	--	--	0.2	--	
Engine Volume, ft^3		2.16	3.76	--	--	2.0	--	
Specific Transmission Volume, ft^3/hp(s)		0.0	0.005	0.02	--	--	--	
Transmission Volume, ft^3		0.0	0.09	0.40	Volume of batteries, motors, and controls			
Specific Volume of Conventional Fuel, ft^3/hp(t)-hr		0.0029	0.0029	0.25[(d)]	0.0029	0.0029	0.0029	

TABLE B-2. (Continued)

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter(a)	Thermionic Converter(a)	Thermo-photo-voltaic Converter(a)	Magneto-hydrodynamics and Nuclear Devices
Energy Storage Volume, ft^3		0.19	0.15	3.35	0.43	0.43	0.34	Found to be unsuitable for this application - see discussion.
Engine, Transmission, and Conventional Fuel Volume, ft^3	--	2.4	4.0	3.8	Similar to Thermionic	2.4+	Similar to Thermionic	
Specific Engine Cost, $/hp(s) or hp(e)		6.0	7.0	--	Not known but is expected to be $100/hp(e) or higher			
Engine Cost, $		80	130	--	--	--	--	
Specific Transmission Cost, $/hp(s)		0.0	1.0	3.0	--	--	--	
Transmission Cost, $		0.0	19	60	Cost of batteries, motors, and controls			
Specific Energy Container Cost, $/hp(t)-hr		Nominal	Nominal	100(e)	Nominal	Nominal	Nominal	
Energy Container Cost, $		--	--	1340	--	--	--	
Engine, Transmission, and Energy Container Cost, $	--	80	150	1400	Very high			

(a) Considered to be part of a hybrid system. Their power rating is based on the cruise power requirement which is one-half the maximum power. The weight, size, and cost of associated batteries, motors, and controls would have to be added to the values listed in this table for the converters alone.

(b) Value given in hp(s)-hr.

(c) Value given in lb/hp(s)-hr.

(d) Value given in ft^3/hp(s)-hr.

(e) Value given in $/hp(s)-hr.

TABLE B-3. ESTIMATED PROPULSION SYSTEM CHARACTERISTICS FOR DELIVERY VAN APPLICATION

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter[a]	Thermionic Converter[a]	Thermo-photo-voltaic Converter[a]	Magneto-hydrodynamics and Nuclear Devices
Maximum Delivered Power, hp(s)	96	96	96	96	96	96	96	Found to be unsuitable for this application – see discussion.
Assumed Transmission and Drive Efficiency, %	90	95	85	80	80	80	80	
Maximum Engine Power, hp(s) or hp(e)	107	101	113	120	120	120	120	
Rated Engine Power, hp(s) or hp(e)	107	81	113	120	60	60	60	
Specific Engine Weight, lb/hp(s) or hp(e)	3.3	5.6	8.4	--	13	10	13	
Engine Weight, lb	350	450	950	--	780	600	780	
Specific Transmission Weight, lb/hp(s)	0.5	0.0	0.7	2.0	--	--	--	
Transmission Weight, lb	54	0.0	79	240	Weight of battery, motors, and controls			
Energy Delivered to Vehicle, hp(s)-hr	60	60	60	60	60	60	60	
Energy Delivered by Engine, hp(s)-hr or hp(e)-hr	67	63	71	--	75	75	75	
Peak Engine Efficiency, %	33	29	39	--	10	10	13	
Assumed Average Efficiency, %	23	20	27.5	80	9	9	11.5	
Required Stored Energy, hp(t)-hr	290	315	257	75[b]	830	830	650	
Specific Weight of Conventional Fuel, lb/hp(t)-hr	0.15	0.15	0.15	100[c]	0.15	0.15	0.15	
Energy Storage Weight, lb	44	47	39	7500	120	120	98	
Engine, Transmission, and Conventional Fuel Weight, lb	450	500	1100	7700	900+	720+	880+	
Specific Engine Volume, ft^3/hp(s) or hp(e)	0.10	0.13	0.16	--	--	0.2	--	
Engine Volume, ft^3	10.7	10.5	18.1	--	--	12.0	--	
Specific Transmission Volume, ft^3/hp(s)	0.004	0.0	0.005	0.02	--	--	--	
Transmission Volume, ft^3	0.43	0.0	0.56	2.40	Volume of batteries, motors, and controls			
Specific Volume of Conventional Fuel, ft^3/hp(t)-hr	0.0029	0.0029	0.0029	0.25[d]	0.0029	0.0029	0.0029	
Energy Storage Volume, ft^3	0.84	0.91	0.74	18.80	2.40	2.40	1.88	

TABLE B-3. (Continued)

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter(a)	Thermionic Converter(a)	Thermo-photo-voltaic Converter(a)	Magneto-hydrodynamics and Nuclear Devices
Engine, Transmission, and Conventional Fuel Volume, ft^3	12	11	19	21	Similar to Thermionic	14+	Similar to Thermionic	Found to be unsuitable for this application - see discussion.
Specific Engine Cost, \$/hp(s) or hp(e)	3.4	4.4	5.4	--	Not known but is expected to be \$100/hp(e) or higher			
Engine Cost, \$	360	360	610	--	--	--	--	
Specific Transmission Cost, \$/hp(s)	0.4	0.0	1.0	3.0	--	--	--	
Transmission Cost, \$	43	0.0	110	360	Cost of batteries, motors, and controls			
Specific Energy Container Cost, \$/hp(t)-hr	Nominal	Nominal	Nominal	100(e)	Nominal	Nominal	Nominal	
Energy Container Cost, \$	--	--	--	7500	--	--	--	
Engine, Transmission, and Energy Container Cost, \$	400	360	720	7900	Very high			

(a) Considered to be part of a hybrid system. Their power rating is based on the cruise power requirement which is one-half the maximum power. The weight, size, and cost of associated batteries, motors, and controls would have to be added to the values listed in this table for the converters alone.

(b) Value given in hp(s)-hr.

(c) Value given in lb/hp(s)-hr.

(d) Value given in ft^3/hp(s)-hr.

(e) Value given in \$/hp(s)-hr.

TABLE B-4. ESTIMATED PROPULSION SYSTEM CHARACTERISTICS FOR CITY TAXI APPLICATION

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter[(a)]	Thermionic Converter[(a)]	Thermo-photo-voltaic Converter[(a)]	Magneto-hydrodynamics and Nuclear Devices
Maximum Delivered Power, hp(s)	48	48	48	48	48	48	48	Found to be unsuitable for this application – see discussion.
Assumed Transmission and Drive Efficiency, %	90	95	85	80	80	80	80	
Maximum Engine Power, hp(s) or hp(e)	53.4	50.5	57.5	60	60	60	60	
Rated Engine Power, hp(s) or hp(e)	53.4	40.4	57.5	60	30	30	30	
Specific Engine Weight, lb/hp(s) or hp(e)	3.7	6.4	9.0	--	13	10	13	
Engine Weight, lb	200	260	520	--	390	300	390	
Specific Transmission Weight, lb/hp(s)	0.5	0.0	0.7	2.0	--	--	--	
Transmission Weight, lb	27	0.0	40	120	Weight of battery, motors, and controls			
Energy Delivered to Vehicle, hp(s)-hr	100	100	100	100	100	100	100	
Energy Delivered by Engine, hp(s)-hr or hp(e)-hr	111	105	118	--	125	125	125	
Peak Engine Efficiency, %	31	27	37	--	10	10	13	
Assumed Average Efficiency, %	21.5	19	26	80	9	9	11.5	
Required Stored Energy, hp(t)-hr	516	553	455	125[(b)]	1390	1390	1090	
Specific Weight of Conventional Fuel, lb/hp(t)-hr	0.15	0.15	0.15	100[(c)]	0.15	0.15	0.15	
Energy Storage Weight, lb	77	83	68	12,500	210	210	160	
Engine, Transmission, and Conventional Fuel Weight, lb	300	340	630	13,000	600+	510+	550+	
Specific Engine Volume, ft^3/hp(s) or hp(e)	0.11	0.14	0.19	--	--	0.2	--	
Engine Volume, ft^3	5.87	5.60	10.90	--	--	6.0	--	
Specific Transmission Volume, ft^3/hp(s)	0.004	0.0	0.005	0.02	--	--	--	
Transmission Volume, ft^3	0.21	0.0	0.29	1.20	Volume of batteries, motors, and controls			
Specific Volume of Conventional Fuel, ft^3/hp(t)-hr	0.0029	0.0029	0.0029	0.25[(d)]	0.0029	0.0029	0.0029	
Energy Storage Volume, ft^3	1.50	1.60	1.32	31.0	4.04	4.04	3.[illegible]	

TABLE B-4. (Continued)

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter(a)	Thermonic Converter(a)	Thermo-photo-voltaic Converter(a)	Magneto-hydrodynamics and Nuclear Devices
Engine, Transmission, and Conventional Fuel Volume, ft^3	7.6	7.2	12	32	Similar to Thermionic	10+	Similar to Thermionic	Found to be unsuitable for this application – see discussion.
Specific Engine Cost, \$/hp(s) or hp(e)	4.0	5.0	6.0	--	Not known but is expected to be \$100/hp(e) or higher			
Engine Cost, \$	210	200	340	--	--	--	--	
Specific Transmission Cost, \$/hp(s)	0.4	0.0	1.0	3.0	--	--	--	
Transmission Cost, \$	21	0.0	58	180	Cost of batteries, motors, and controls			
Specific Energy Container Cost, \$/hp(t)-hr	Nominal	Nominal	Nominal	100(e)	Nominal	Nominal	Nominal	
Energy Container Cost, \$	--	--	--	12,500	--	--	--	
Engine, Transmission, and Energy Container Cost, \$	230	200	400	13,000	Very high			

(a) Considered to be part of a hybrid system. Their power rating is based on the cruise power requirement which is one-half the maximum power. The weight, size, and cost of associated batteries, motors, and controls would have to be added to the values listed in this table for the converters alone.

(b) Value given in hp(s)-hr.

(c) Value given in lb/hp(s)-hr.

(d) Value given in ft^3/hp(s)-hr.

(e) Value given in \$/hp(s)-hr.

TABLE B-5. ESTIMATED PROPULSION SYSTEM CHARACTERISTICS FOR CITY BUS APPLICATION

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter[a]	Thermionic Converter[a]	Thermo-photo-voltaic Converter[a]	Magneto-hydrodynamics and Nuclear Devices
Maximum Delivered Power, hp(s)	180	180	180	180	180	180	180	Found to be unsuitable for this application - see discussion.
Assumed Transmission and Drive Efficiency, %	90	90	85	80	80	80	80	
Maximum Engine Power, hp(s) or hp(e)	200	200	212	225	225	225	225	
Rated Engine Power, hp(s) or hp(e)	200	160	212	225	112	112	112	
Specific Engine Weight, lb/hp(s) or hp(e)	4.0	10.0	16.0	--	17	13	17	
Engine Weight, lb	800	1600.0	3390	--	1900	1460	1900	
Specific Transmission Weight, lb/hp(s)	2.0	2.0	3.0	3.0	--	--	--	
Transmission Weight, lb	400	320	640	680	Weight of battery, motors, and controls			
Energy Delivered to Vehicle, hp(s)-hr	400	400	400	400	400	400	400	
Energy Delivered by Engine, hp(s)-hr or hp(e)-hr	445	445	470	--	500	500	500	
Peak Engine Efficiency, %	35	30	40	--	10	10	13	
Assumed Average Efficiency, %	24.5	21	28	80	9	9	11.5	
Required Stored Energy, hp(t)-hr	1820	2120	1680	500[b]	5560	5560	4340	
Specific Weight of Conventional Fuel, lb/hp(t)-hr	0.15	0.15	0.15	100[c]	0.15	0.15	0.15	
Energy Storage Weight, lb	270	320	250	50,000	830	830	650	
Engine, Transmission, and Conventional Fuel Weight, lb	1500	2200	4300	51,000	2700+	2300+	2600+	
Specific Engine Volume, ft^3/hp(s) or hp(e)	0.12	0.24	0.30	--	--	0.27	--	
Engine Volume, ft^3	24.0	36.5	63.6	--	--	30.2	--	
Specific Transmission Volume, ft^3/hp(s)	0.02	0.02	0.03	0.03	--	--	--	
Transmission Volume, ft^3	4.0	3.04	6.36	6.75	Volume of batteries, motors, and controls			
Specific Volume of Conventional Fuel, ft^3/hp(t)-hr	0.0029	0.0029	0.0029	0.25[d]	0.0029	0.0029	0.0029	
Energy Storage Volume, ft^3	5.28	6.15	4.86	125	16.1	16.1	12.6	

TABLE B-5. (Continued)

Parameters	Gas Turbine	Rankine Engine	Stirling Engine	Flywheel Energy Storage	Thermo-electric Converter[(a)]	Thermionic Converter[(a)]	Thermo-photo-voltaic Converter[(a)]	Magneto-hydrodynamics and Nuclear Devices
Engine, Transmission, and Conventional Fuel Volume, ft^3	33	46	75	130	Similar to Thermionic	46+	Similar to Thermionic	
Specific Engine Cost, \$/hp(s) or hp(e)				--	Not known but is expected to be \$100/hp(e) or higher			Found to be unsuitable for this application – see discussion.
Engine Cost, \$				--	--	--	--	
Specific Transmission Cost, \$/hp(s)				14.0	--	--	--	
Transmission Cost, \$				3200	Cost of batteries, motors, and controls			
Specific Energy Container Cost, \$/hp(t)-hr				100[(e)]	Nominal	Nominal	Nominal	
Energy Container Cost, \$				50,000	--	--	--	
Engine, Transmission, and Energy Container Cost, \$	Should be similar to or somewhat higher than cost of commercial-vehicle diesel engine and hydrokinetic transmission [i.e., \$25 to 35/hp(s)]			53,000		Very high		

(a) Considered to be part of a hybrid system. Their power rating is based on the cruise power requirement which is one-half the maximum power. The weight, size, and cost of associated batteries, motors, and controls would have to be added to the values listed in this table for the converters alone.

(b) Value given in hp(s)-hr.

(c) Value given in lb/hp(s)-hr.

(d) Value given in ft^3/hp(s)-hr.

(e) Value given in \$/hp(s)-hr.

APPENDIX C

REFERENCES

APPENDIX C

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